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Boiling Heat-Transfer Processes and Their Application in the Cooling of High Heat Flux Devices

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Water cooling is used extensively at AEDC to prevent the failure of high-enthalpy arc heater components at elevated heat loads where cooling by radiation, conduction, and natural convection from air circulation are ineffective. Other cooling techniques such as the use of liquid metals, transpiration, or film cooling pose their own disadvantages or limitations and are certainly possibilities for providing cooling at much more severe heating conditions where water cooling alone may be inadequate. The current interest is in exploiting backside water cooling to the limit of its capability, especially for arc heater nozzles. Such an application requires an understanding of the parameters that affect the cooling processes and the determination of the limiting point where the surface fails due to inadequate cooling. This report reviews the different processes encountered in backside water cooling and identifies those processes that are desirable for effective cooling of high-enthalpy facility components; identifies those parameters which affect the cooling processes of interest and the trend of cooling performance when the parameters are individually varied; and presents a summary of applicable theoretical, analytical, and experimental work in surface cooling performed in the past 50 years.

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PREFACE

The work reported herein was performed by Arnold Engineering Development Center (AEDC), Air Force Materiel Command (AFMC), under Program Element 65807F. The Air Force Program Managers were Capt. D. G. Burgess, Capt. H. Martin, and Lt. P. Zeman, DOT. The work was performed by Calspan Corporation/AEDC Operations, technical services contractor for the Aerospace Flight Dynamics testing effort at AEDC, AFMC, Arnold Air Force Base, TN. The work was performed in the Aerospace Systems Facility (ASF) and the Technology and Development Facility (TDF) under AEDC Project Number DD01, Job Number 0115. The work was conducted in conjunction with the development of the High Temperature Wall Laboratory (HTWL) during the period between 1 October 1990 and 16 October 1992. The manuscript was submitted for publication on February 25, 1993.

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1.0 INTRODUCTION

The work reported herein was performed by Arnold Engineering Development Center (AEDC), Air Force Materiel Command (AFMC), under Program Element 65807F. The Air Force Program Managers were Capt. D. G. Burgess, Capt. H. Martin, and Lt. P. Zeman, DOT. The work was performed by Calspan Corporation/AEDC Operations, technical services contractor for the Aerospace Flight Dynamics testing effort at AEDC, AFMC, Arnold Air Force Base, TN. The work was performed in the Aerospace Systems Facility (ASF) and the Technology and Development Facility (TDF) under AEDC Project Number DD01, Job Number 0115. The work was conducted during the period between 1 October 1990 and 16 October 1992.

Water cooling is used extensively at AEDC to prevent the failure of high-enthalpy arc heater components at elevated heat loads where cooling by radiation, conduction, and natural convection from air circulation are ineffective. Other cooling techniques such as the use of liquid metals, transpiration or film cooling pose their own set of disadvantages or limitations (Ref. 1) but are certainly possibilities for providing cooling at much more severe heating conditions where water cooling alone may prove to be inadequate. The current interest is in exploiting backside water cooling to the limit of its capability, especially for arc heater nozzles. Such an application requires an understanding of the parameters that affect the cooling processes and the determination of the limiting point* where the surface fails due to inadequate cooling.

The purpose of this report is threefold: (1) to review the different processes encountered in backside water cooling and identify those processes that are desirable for effective cooling of high-enthalpy facility components; (2) to identify those parameters which affect the cooling processes of interest and the trend of cooling performance when the parameters are individually varied; and (3) to present a summary of applicable theoretical, analytical, and experimental work in surface cooling performed in the past 50 years.

2.0 THE BASIC BOILING CURVE

The characteristic temperature response of a surface cooled by an adjacent fluid as heat flux is increased was first proposed by Nukiyama (Ref. 2) in 1934. Figure 1 shows the typical

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* The limiting point is also identified as the burnout point, boiling crisis, departure from nucleate boiling (DNB), or critical heat flux (CHF) condition, among other terms by various authors. Terminology for the limiting point will be discussed in Section 2.0.

shape of the curve, known as the boiling curve, obtained when heat flux from a cooled surface is plotted as a function of the liquid superheat (the difference between the surface temperature and the liquid saturation temperature). At low heat flux (segment A-B in Fig. 1), the surface temperature is below the saturation temperature of water and similar liquids at or above atmospheric pressure. Cooling at low heat flux is by pure convection heat transfer and is fairly well understood. In this region, analytical solutions and experimental correlations (for more complex flow geometries) have been derived which allow for accurate prediction of heat transfer in various cooling configurations (see, for example, Chapters 8-9 of Lienhard (Ref. 3), Chapter 6 of White (Ref. 4), or Kays and Crawford (Ref. 5), Chapters 13-14). As heat flux from the surface is increased beyond the point where the surface temperature becomes significantly greater** than the water saturation temperature (point B in Fig. 1), vapor generation, better known as boiling, occurs from discrete nucleation sites on the surface or in the bulk fluid depending on the bulk fluid temperature. The region of boiling from discrete sites is termed nucleate boiling and is shown as segment B-C in Fig. 1.

As the upper limit of nucleate boiling is reached (point C in Fig. 1), a typically undesirable transition to another form of boiling, known as film boiling, begins. Because the boiling near the limit of nucleate boiling becomes so violent, numerous vapor bubbles begin to combine near the surface. Since the vapor has a much lower thermal conductivity than the liquid, the patches of vapor effectively insulate the surface from the efficient heat transfer to the liquid as experienced during nucleate boiling. Unlike a temperature-driven system which would proceed toward point D† in Fig. 1, a heat flux-driven system such as a high-enthalpy arc facility would proceed directly toward point E††. The processes involved when a boiling system undergoes the path from the limit of nucleate boiling (point C) toward stable film boiling (point E) are highly unstable, and failure of the surface due to melting occurs in most cases because the melting temperature of the surface material is lower than the temperature associated with stable film boiling at point E. The point at which transition to film boiling begins is often termed the burnout point, even though, in some cases, failure does not occur (e.g., a temperature-driven system or a surface material with a very high melt temperature).

** A certain degree of liquid superheat is required to establish a flow of heat to create vapor bubbles. The superheat can become quite large for very smooth surfaces or highly wettable fluids (e.g., refrigerants or liquid metals) which can lead to explosive boiling (known as bumping) that can lead to structural damage to hardware.

† Transition boiling occurs between the limit of nucleate boiling (point C) and point D, which is known as the Leidenfrost point. The path between the points is dependent on numerous factors and the heat flux becomes a multivalued function of the liquid superheat (e.g., see Witte and Lienhard, Ref. 6). This region is inaccessible with a heat flux controlled system.

†† Point E lies on a portion of the boiling curve known as film boiling. Beginning at the Leidenfrost point (point D), heat transfer across the vapor film adjacent to the surface increases due to radiation from the wall, through the vapor, to the bulk fluid.

The burnout point is known also as the departure from nucleate boiling (DNB), fast burnout, and burnout of the first kind in systems which involve a subcooled* liquid. In saturated liquid boiling systems, another type of burnout occurs (usually at lower heat flux than subcooled boiling), and the terms slow burnout, dryout, or burnout of the second kind are used (see Ref. 8). Boiling crisis and critical heat flux condition are used in a general sense to describe all burnout situations. Because of a lack of standard nomenclature, hereafter the point at which transition to film boiling begins is designated the burnout point or boiling crisis, and the peak, ultimate, or maximum heat flux is termed the critical heat flux (CHF), except where designated differently in the referenced material.

Design of the cooled components of a high-enthalpy facility for safe operation necessitates the cooling processes to lie to the left of, and well below, point C in Fig. 1. High heat-transfer coefficients, hence, efficient energy transfer, can be realized when operating in the nucleate boiling regime rather than the pure convection regime at low-to-moderate coolant pressure and velocity. Any method used to increase the CHF will provide a factor of safety when operating well into the fully developed nucleate boiling regime. The determination of the CHF, along with the shape of the nucleate boiling curve for a given cooling system, remains one of the most challenging areas of research despite the considerable attention it has received in the last 50 years (Ref. 9). Complexity of the boiling mechanisms, their unsteady nature, and the small scale of the heat-transfer processes have hindered the development of theoretical models and understanding of the phenomena from experimental studies. Moreover, the cooling flow structure and, hence, the heat-transfer processes vary depending on the heating condition, fluid properties, and surface geometry. For instance, boiling heat transfer that occurs in a high-enthalpy facility (subcooled flow, surface boiling optimized for effective wall cooling over short lengths at high heat loads) can be quite different than that which occurs in a nuclear reactor (saturated flow with possible slug or annular two-phase flow, bulk boiling optimized for efficient energy transfer over great lengths at low-to-moderate heat loads).

The objective of a large portion of the experimental work undertaken in recent years in the area of boiling heat transfer has been to obtain an empirical or semiempirical correlation for a given cooling configuration and set of test conditions. Many of the correlations have limited application and cannot be generalized for use in analyzing other cooling configurations and conditions. The experiments do provide a qualitative evaluation of what parameters affect the boiling processes of interest and the trend in cooling performance when each of the

* If the cooling fluid bulk temperature is well below the saturation temperature of the fluid, then the fluid is known as subcooled. Boiling that occurs on a surface adjacent to such a fluid is known as subcooled local or surface boiling. If the bulk temperature is at or near the saturation temperature, the fluid is known as saturated. Bulk boiling can occur when the fluid is near saturation, and undergoes various flow regimes in forced flow such as slug, annular, and drop flow. See Collier (Ref. 7) for further discussion of saturated flows.

parameters is varied. Optimizing the influential parameters for high-enthalpy facility component cooling will, in turn, aid in identifying those boiling mechanisms which play a part in the cooling process of interest. Once the mechanisms are identified, appropriate analytical models and experimental data may be assembled.

3.0 PARAMETRIC EFFECTS ON THE BOILING CURVE

More than 20 parameters that affect the cooling process have been identified. Some parameters play an important role from pure convection to boiling crisis while others play intermittent roles. For instance, velocity has a large effect on cooling performance in the pure convection regime and at boiling crisis, but it has virtually no effect in the nucleate boiling regime. Therefore, the contribution of each parameter in each of the three regions of interest (pure convection, nucleate boiling, boiling crisis) will be discussed. The effects of various parameters on the microscopic boiling scale (i.e., bubble characteristics) have not been studied in great detail and are presented here for the more influential parameters. The cooling process is not without parametric distortion; that is, the change in one parameter causes a change in another parameter. Because of the complexity of the boiling process, many of the parametric distortions are ill-defined or unknown; therefore, a discussion of these distortions included here is limited. Several excellent reviews of various parametric effects on boiling are available and are drawn upon in the following review, most notably: boiling in general—Tong (Ref. 10), Westwater (Refs. 11 and 12), Rohsenow (Refs. 13 and 14), Addoms (Ref. 15), and Katto (Ref. 16); convective boiling—Collier (Ref. 7); subcooled flow boiling—Boyd (Refs. 1 and 17), Bergles (Ref. 8), Macbeth (Ref. 18), Hewitt (Ref. 19), Fiori and Bergles (Ref. 20), and Hughes (Ref. 21).

3.1 IMPORTANT PARAMETERS AND THEIR EFFECTS ON HEAT TRANSFER

Velocity

Velocity (or mass velocity, G, when combined with the fluid density), as mentioned previously, plays an intermittent role in the cooling process. Figure 2 (from Kreith and Summerfield, Ref. 22) illustrates the significant effect of fluid (butanol) velocity as heat flux from an adjacent surface increases in pure convection heat transfer. However, as transition to boiling occurs, the effect of velocity diminishes. Clark and Rohsenow (Ref. 23) and Tong (Ref. 10), among others, have noted that velocity has little or no effect on heat transfer in vigorous (or fully developed) nucleate boiling. Analysis of boiling processes is divided into two categories with regard to velocity. Boiling which occurs on a surface or in a fluid initially at rest where only free-convection motion is present is known as pool boiling. Boiling which occurs on a surface or in a fluid where the fluid has a bulk (imposed by some external source such as a pump) and free-convection motion is known as forced-convection boiling or simply flow boiling. The advantages of flow boiling over pool boiling, when present in a cooling process, become evident as boiling crisis is approached. McAdams, et al. (Ref. 24) showed that the CHF could be increased with increasing velocity (Fig. 3). Grace (Ref. 25) generalized the velocity effect on CHF for various qualities of water liquid/vapor systems as shown in Fig. 4. An inversion in velocity effectiveness occurs at higher vapor qualities (i.e., saturated fluid) as illustrated in Fig. 4; however, as will be discussed next, subcooling (vapor qualities less than zero) provides the largest values of CHF, where increasing mass velocity increases the CHF for a given degree of subcooling (Fig. 4).

Analyzing the boiling process on a microscopic scale, Abdelmessih, et al. (Ref. 26) found that both the maximum bubble radius and the bubble lifespan decrease as the liquid velocity increases (Fig. 5). Gunther (Ref. 27) found, in addition to decreasing the bubble radius and lifetime, that increasing velocity also decreases the bubble population. Tippets (Ref. 28) found that when high velocity is combined with large subcoolings, the flow adjacent to a surface sustaining boiling is an irregular, frothy layer made up of bubbles that grow, slide, and then collapse along the surface. These bubble characteristics make their detection and measurement at high flow velocity difficult.

Subcooling

Subcooling, or the amount that the fluid bulk temperature is below the fluid saturation temperature, has been shown to have an effect similar to velocity on the cooling process. Researchers have correlated experimental data using a fluid bulk temperature measured at various locations in a given apparatus. Generally, for larger length-to-diameter ratios where the bulk fluid may enter at subcooled conditions but reach saturation conditions somewhere along the heated length, the inlet subcooling (inlet saturation temperature minus the inlet bulk temperature) is typically used in analyses. For constant diameter configurations and uniform heating where burnout occurs at or near the exit of the heater, the exit subcooling (exit saturation temperature minus the exit bulk temperature) is typically used. Some researchers have used a mean average bulk temperature (average of the inlet and outlet bulk temperatures) in some instances to define subcooling. In some cases, subcooling is calculated using an inlet bulk temperature and an exit pressure (exit saturation temperature). For a configuration such as an arc heater nozzle where nonuniform heating, velocity, and pressure exist along the length of the nozzle, a local subcooling using a local bulk temperature and saturation temperature is more appropriate. A discussion of the general subcooling effects follows, with an effort made to identify the type of subcooling used in the experimental results cited.

McAdams, et al. (Ref. 24) showed that inlet subcooling has a slight effect on heat transfer in the pure convection and the nucleate boiling regions of the boiling curve because of small variations in physical properties of the fluid at different temperatures. Subcooling, like velocity, has its primary advantages at boiling crisis. Van Huff and Rousar (Ref. 29) and Kreith (Ref. 30) indicated that subcooling and velocity are the most important variables on burnout (CHF). Green, et al. (Ref. 31) measured burnout heat flux for vertical water flow in a rectangular channel at high pressure and found that increasing both velocity and inlet subcooling increased the burnout heat flux (Fig. 6). Van Huff and Rousar (Ref. 29) combined velocity and an average subcooling and derived a linear fit to previous data (Fig. 7). Macbeth (Ref. 18) found that increasing inlet subcooling increased CHF linearly at constant mass velocity; however, Katto and Yokoya (Ref. 32) indicated that nonlinear regimes can occur at high mass velocity at low-to-moderate inlet subcooling, and Lienhard (Ref. 33) found that the maximum heat flux (CHF) versus degree of subcooling curve "flattens" out for any velocity above a threshold degree of subcooling. In any event, it is likely that the CHF generally can be increased by increasing velocity and subcooling.

On a microscopic scale, Gunther's work (Ref. 27) with a flow boiling apparatus found that the bubble lifetime, surface coverage, and maximum radius decreased with increased exit subcooling, while the bubble population, which initially decreases, increases for subcoolings above 72°C (130°F as shown in Fig. 8). As reported by Westwater (Ref. 12), the motion pictures of Dew (Ref. 34) confirmed Gunther's findings except bubble population, which decreased with increased average subcooling. Ibrahim and Judd (Ref. 35) confirmed that the growth period decreased with increased subcooling and, in addition, found that the waiting period between bubble generations from an active nucleation site also decreased with increased subcooling except at low subcooling (below 22°C). It should be noted, however, that Ibrahim and Judd performed their subcooling studies on a pool boiling apparatus.

Pressure

The primary contribution of pressure on a cooling process is its effect on the saturation temperature of the fluid, although an interesting inversion of pressure effect occurs at boiling crisis. Kreith and Summerfield (Ref. 22), who studied flow boiling with butanol, found that increasing pressure had no effect on heat transfer in the pure convection region, but increased wall temperature for a given heat flux in the nucleate boiling region, as shown in Fig. 2. An increase in wall temperature is not particularly beneficial; however, some improvement to the CHF can be achieved by increasing pressure - up to a point. Cichelli and Bonilla (Ref. 36), while studying pool boiling of organic liquids, found that heat flux increased with increasing pressure to a maximum value which occurred at about one-third of the critical pressure (Fig. 9). Addoms (Ref. 15) obtained similar results from pool boiling of water; however, the maximum peak heat flux varied with heater wire size (Fig. 10). Chang (Ref. 37) suggested a similar maximum CHF in flow boiling that occurs at various pressures between 500 and 1,000 psia, depending on mass velocity and subcooling. Boyd (Ref. 1) suggested a similar "optimum pressure" as shown in Fig. 11.

Tolubinsky and Kostanchuk (Ref. 38) found that increased pressure decreased the maximum bubble diameter and the generation frequency (Fig. 12), although the former appears to remain constant or to begin increasing above a pressure of 8 to 10 bar (120 to 150 psia). The bubble life span is inversely related to the generation frequency, and, therefore, increases with increased pressure.

Surface Roughness

Surface roughness plays a complicated role in cooling efficiency and is divided into micro-roughness (typically smaller than 3 μm) and macro-roughness (artificial protrusions such as knurling, fins, or ribs). Corty and Foust (Ref. 39) performed an extensive micro-roughness study of various fluids in pool boiling on copper and nickel surfaces. They found that increased roughness had no effect in the pure convection region, but heat-transfer coefficients were more than doubled in the nucleate boiling region for the roughest surface when compared to the smoothest surface (Fig. 13). Brown (Ref. 40), however, found a modest effect of surface finish in subcooled, low-velocity flow boiling at moderate heat flux (Fig. 14). Berenson (Ref. 41) and later Ramilison and Lienhard (Ref. 42) found a modest surface finish dependence of boiling burnout (up to approximately 20-percent decrease from smooth to rough surface) in pool boiling. Other than Weatherhead (Ref. 43), who concluded that CHF is decreased for very smooth surfaces, other investigators such as Leung, et al. (Ref. 44), Aladyev, et al. (Ref. 45), and Bergles and Morton (Ref. 46) found little or no effect of surface micro-roughness on CHF in flow boiling. Bergles and Morton further determined that any improvements of the burnout heat flux caused by roughness appear to diminish with increasing velocity and subcooling.

Macro-roughness has been found to have a more pronounced effect on forced flow cooling efficiency than the micro-roughness previously discussed. Types of macro-roughness used in forced flow include knurling (Durant, et al., Ref. 47), fins (Kovalev, et al., Ref. 48, and Shim, et al., Ref. 49), and ribs (Ravigururajan and Bergles, Ref. 50, and Akhanda and James, Ref. 51). Durant and his team determined that liquid film heat-transfer coefficients of knurled surfaces were up to 75-percent higher than those of smooth surfaces. Burnout heat flux was shown to increase by as much as 80 percent (Fig. 15). The types of roughness investigated by Durant's team were comparatively large (greater than 130 μm) over those discussed in the previous paragraph. Macro-roughness is considered a heat-transfer enhancement technique rather than a basic parameter affecting heat transfer that would be inherent in a given system. An excellent review of this enhancement technique is presented by Webb (Ref. 52); however, most of the surface geometries have been developed for pool boiling only.

Coolant Properties

The understanding of the effects of coolant properties on the heat-transfer processes in boiling remains limited, even though selection of the fluid is a most important consideration in boiling applications (Ref. 1). The physical properties of a given fluid such as liquid/vapor density ratio, latent heat of vaporization, surface tension, heat capacity, and transport properties (i.e., viscosity and thermal conductivity) may play a role in the cooling process; however, distinguishing their individual contribution is difficult, and little or no data exist to ascertain their effect. The properties are related to the pressure and/or the temperature of the fluid, and the problem of parametric distortion discussed earlier becomes evident.

The limited studies of the microscale mechanisms during boiling have produced conflicting information as to the effect of latent heat. A number of investigations (Refs. 53-56) have revealed that latent heat transport accounts for as much as 50 percent of the measured heat flux in subcooled boiling; however, Forster and Grief (Ref. 57), Brown (Ref. 40), and Plesset and Prosperetti (Ref. 58) found very little latent heat contribution.

In their work with a pool boiling apparatus, Pike, et al. (Ref. 59) found that increasing viscosity hindered the onset of boiling (decreased heat-transfer coefficient, increased wall superheat) and lowered the CHF. As will be discussed under transient operation effects, it has been proposed that transient operation is viscosity-dependent, thereby contributing to an observed difference in CHF for the transient and steady-state heating of certain fluids.

Surface tension may or may not play a beneficial role in the cooling process (Ref. 11). Fiori and Bergles (Ref. 20) identified several investigations that clearly showed that CHF decreases with decreasing surface tension. The addition of various amounts of surface-active agent has been found to increase the heat transfer quite significantly at the expense of reducing the CHF. For example, Frost and Kippenhan (Ref. 54) showed that for a given wall temperature, the heat flux increased by approximately 50 percent when a surface-active agent was added to water, thereby reducing the surface tension (Fig. 16). The reduction of surface tension (or increased wettability) had a detrimental (decreasing) effect on the CHF, which agrees with previous results, and is evident in Fig. 16. However, a recent study by Sadasivan and Lienhard (Ref. 60) showed an increased CHF with the addition of the surfactant sodium dodecyl sulfate to water. They also noted that bubble diameters were smaller, and the number of bubbles was much greater with the surfactant present.

Improvements to the CHF have been demonstrated in a limited number of investigations using binary mixture components. Papaioannou and Koumoutsos (Ref. 61) and Wei and Maa (Ref. 62) found that polymeric additives to water increased heat-transfer coefficients and broadened the peak (CHF) of the boiling curve. Tolubinskiy and Matorin (Ref. 63) found that for ethanol-water and acetone-water mixtures, the maximum CHF (occurring at 20-percent ethanol/80-percent water and 10-percent acetone/90-percent water) exceeded the CHF for water by 30 and 40 percent, respectively (Fig. 17).

A number of coolants such as refrigerants, cryogenics, organics, binary mixtures, liquid metals, and propellants, in addition to water, have been investigated for various cooling applications. A propellant is probably not attractive for use as a coolant in a high-enthalpy facility, except in specific test article configurations. The use of organics is unlikely, except possibly as a trace additive or in a binary mixture, because of their lower performance compared to that of water (Ref. 64) and their tendency to decompose at surface temperatures experienced at nucleate boiling heat flux levels (Ref. 8). Binary mixtures, as discussed previously, may be an attractive approach to improve the CHF in a water-cooled system. Liquid metals may be considered in the future as an alternative or complement to water cooling. Because of their high thermal conductivity, liquid metals have proved useful in convective cooling applications. Boiling in liquid metals, however, is somewhat confused. Lyon, et al. (Ref. 65) showed that mercury, for instance, appears to enter film boiling at surprisingly low wall superheats. And because of the high wettability of liquid metals, high superheats are normally required to initiate boiling, in some cases having an explosive transition that can cause structural damage to hardware. A thorough understanding of boiling metals is needed before they can be applied in high heat flux/cooling situations. Refrigerants such as Freon® have been used in previous experiments (Refs. 66-69) to simulate water cooling at high heat flux conditions since the CHF can be achieved at much lower heat flux. Scaling methods are applied to the simulation to achieve high heat flux/water cooling predictions (see, for example, Ahmad, Ref. 70). A good example of this "fluid-to-fluid" scaling is presented by Bergles in Ref. 8. Use of refrigerants for actual cooling in high-enthalpy facilities is not recommended because of their low thermal conductivity and CHF levels. In addition, refrigerants suffer from a high wettability problem not unlike that of liquid metals. Similarly, Powell (Ref. 71) found that low heat-transfer coefficients and CHF levels existed for cryogenics such as hydrogen, nitrogen, and oxygen which would limit the use of liquified gases as a heat-transfer fluid. Refrigerants and cryogenics perform well in systems that have relatively low heat flux requirements and limited space availability.

Wall Material

Although it is considered to be a secondary effect, surface material properties appear to contribute to cooling efficiency. Early experiments in pool boiling demonstrated a difference between stainless steel and copper in nonboiling and boiling water (Ref. 72); copper, gold, and chromium in boiling ethanol (Ref. 73); nickel, tungsten, and Chromel® in boiling water (Ref. 74); and aluminum, copper, iron, and chrome-plated copper in water and various organic liquids (Ref. 75), among others. Later, Magrini and Nannei (Ref. 76) found that the heat-transfer coefficient appears to be dependent on the thermal activity, $\sqrt{\varrho ck}$, for heaters of different metals above a limiting value of thickness (Fig. 18). Kovalev, et al. (Ref. 77) showed that copper with coatings of low thermal conductivity tended to increase the degree of wall superheat over that of clean copper boiling in Freon 113. Cheng and Ragheb (Ref. 78) found that a greater wall superheat was required for an Inconel® surface than a copper surface in flow boiling of water (Fig. 19).

Numerous pool boiling experiments have demonstrated that the CHF has a significant material dependence (Refs. 73-75 and 77) which has not been observed in flow boiling. The data of Cheng and Ragheb presented in Fig. 19 show the CHF to be equivalent for Inconel and copper. Jacket, et al. (Ref. 79) found no effect on CHF due to changing wall material from nickel to Zircaloy® or, later, stainless steel (Ref. 31) in flow boiling of water at high pressure. Van Huff and Rousar (Ref. 29) identified studies in which various metals had no effect on CHF, except where the material and coolant were incompatible (additional reactions occur). Similarly, Vandervort, et al. (Ref. 80) found no difference in CHF for subcooled forced-convection boiling of water on stainless steel 304, stainless steel 316, nickel 200, brass, and Inconel 600. However, Vandervort's team suggested that materials with low values of thermal conductivity, such as stainless steel and Inconel, may exhibit hot spots due to changes in metal crystalline structure when high wall temperature gradients are present (e.g., high heat flux in thin wall tubes). Fiori and Bergles (Ref. 20) suggest that materials with high thermal diffusivity (e.g., aluminum and molybdenum) are expected to have higher CHF.

Wall Thickness

The thickness of the heated wall has been shown to have an effect on the boiling heat flux, but only below a critical or limiting value of wall thickness. Magrini and Nannei (Ref. 76), for instance, found for saturated pool boiling of water that the limiting value was approximately 15 μm (0.0006 in.) for tin and nickel, and negligible for copper and silver. They also determined the limiting value for zinc to be approximately 70 μm (0.0028 in.), although it is not obvious from the data presented in Ref. 76. Del Valle and Kenning (Ref. 81), while studying subcooled water flow at high heat flux, found that thin heated walls (0.08- to 0.2-mm thickness) had no effect in the nonboiling region, but the rate of nucleate boiling heat transfer at a given wall superheat increased with increasing wall thickness (Fig. 20). Tippets (Ref. 28) found that CHF measured for 0.15-mm-thick heater ribbons was approximately 20 percent lower than for 0.25-mm-thick ribbons in saturated water flow. They suggest that thicker walls can maintain a higher internal heat generation rate per unit volume (q''/V) before the critical condition is reached than thin walls, since the latter will have a larger temperature rise at a given heat flux (i.e., q''/V will be greater for a thin-walled tube since the volume is smaller and the heat flux constant). Fiori and Bergles (Ref. 20), using subcooled water in a flow boiling apparatus, showed that thicker-walled tubes (2 mm versus 0.3 mm) had a CHF up to 58 percent higher than thin-walled tubes. Aladyev, et al. (Ref. 44), however, found no effect of wall thickness (0.4 mm to 2 mm) on the CHF for water flow boiling.

Geometry

Geometric considerations that have been experimentally shown to have an effect on the cooling process include the diameter (or hydraulic diameter) of the heater (or channel), heated length (or L/D), the channel cross section and longitudinal configuration, and the flow orientation (Ref. 1). Because of various applications, past experiments associated with studying cooling problems have encompassed a wide range of configurations, and one must be cautious when comparing data from one experiment to another (Ref. 11).

The channel diameter appears to have a varied effect on boiling heat transfer. Schweppe and Foust (Ref. 82) found that for flow boiling with low-velocity water, the wall superheat increased (or heat-transfer coefficient decreased) as channel diameter increased from 11.1 mm to 26.6 mm. Collier (Ref. 7) presented subcooled, flow boiling data of Lee and Obertelli (Ref. 83) and Matzner (Ref. 84) that indicated that CHF increased as channel diameter increased from 5.6 mm to 37.5 mm at constant subcooling (Fig. 21). Bergles (Ref. 85), however, found for subcooled flow boiling that CHF increased as channel diameter decreased from 8.4 mm to 0.6 mm (see Fig. 22), although the percentage increase in CHF is somewhat less when parametric distortion is accounted for (Ref. 20). Similarly, Glushchenko (Ref. 86) found that CHF increased as the tube diameter was reduced below 2 mm. Van Huff and Rousar (Ref. 29) found no effect of channel diameter on CHF for subcooled flow boiling with diameters ranging from 2.7 to 14.9 mm.

Schweppe and Foust (Ref. 82), by varying channel diameter (as previously discussed) and holding the heated length constant, showed for flow boiling with low-velocity water that the wall superheat increased as L/D decreased below 15 to 20. Bergles (Ref. 85) found for flow boiling that CHF increased as L/D was decreased below 35 (Fig. 23). Similarly, Ornatskiy (Ref. 87), using subcooled water flowing in circular ducts, found this limiting L/D to be between 20 to 24, and Jens and Lottes (Ref. 88) showed that, for high-pressure, subcooled flow boiling, changing L/D from 110 to 21 had an increasing effect on CHF. Lee and Obertelli (Ref. 83) found a considerable effect on CHF in subcooled, low-velocity flow for L/D values less than 300 (Fig. 24). Van Huff and Rousar (Ref. 29), however, noted that data acquired by Aerojet General Corporation for high-velocity, highly subcooled water showed very little effect of L/D (ranging from 13 to 100) on CHF. As Boyd (Ref. 1) points out, since the L/D limit is related directly to the flow development, this limit is not a constant and is related to the flow parameters and fluid properties.

The geometric considerations thus far discussed have been primarily for circular tube configurations. External flow over rods (or wires) and rod bundles, jets, internal flow in multiple tube (or tube bundle) configurations, annuli, and, to a lesser extent, rectangular channels have been studied in the past. Internal tube flow and annular flow closely simulate the cooling configurations encountered in a high-enthalpy arc facility. Becker and Hernborg (Ref. 89) found for water flowing in an annulus with inlet subcooling and an inner heated wall (similar to an arc facility nozzle) that CHF data were considerably below those for comparable flow in a circular tube, and Zenkevich, et al. (Ref. 90) also found a difference in CHF between annuli and tubes (at the same length and diameter). Barnett (Ref. 91), however, points out significant difficulties which arise when comparing tube and annulus data and the definition of an appropriate equivalent diameter. In addition, Alekseev, et al. (Ref. 92) found significant heated length effects for annuli at much greater lengths than for circular tubes. Collier (Ref. 7) discusses other characteristics of annular flow of interest. Specifically, eccentricity of the internal heated wall will tend to reduce the CHF (see also Ref. 91). In addition, Collier points out that the outer wall in a concentric annulus system has little or no effect on the CHF for the inner surface.

The channel longitudinal configuration is important to the boiling heat transfer when significant curves or twists are introduced. Such longitudinal changes have an effect on the acceleration and turbulence of the coolant flow. A discussion of these types of configurations is included in the section entitled "Acceleration and Turbulence."

Coolant flow orientation (horizontal, inclined, or vertical) has been shown to have an effect on cooling efficiency due primarily to buoyancy (gravity) causing stratification of the liquid/vapor flow (Refs. 93 and 94). The effect of flow orientation, though, diminishes with increasing subcooling and/or mass velocity, and there appears to be a threshold above which no effect of orientation has been observed (Refs. 1, 7, 94, and 95). Zeigarnik, et al. (Ref. 96), for instance, showed that the orientation of the heater plate in a rectangular flow channel had little effect on the CHF at mass velocities above 4,500 kg/m^2 sec (922 lbm/ft^2 sec) in subcooled water flow (Fig. 25). Boyd, et al. (Ref. 97) suggests that the mass velocity limit is between 3,000 and 4,500 kg/m^2 sec (600 to 900 lbm/ft^2 sec), which agrees with the velocity limit of 4,000 kg/m^2 sec (820 lbm/ft^2 sec) found by Merilo (Ref. 94).

Gas Content

A number of experiments in pool boiling (Refs. 59 and 98) and flow boiling (Refs. 21, 24, and 99) have demonstrated the effect of dissolved gases on pure convection and nucleate boiling heat transfer. McAdams, et al. (Ref. 24), for example, found little effect of dissolved air on forced-convection heating, but the presence of air tended to reduce the wall superheat necessary to initiate and sustain nucleate boiling (Fig. 26) which is typical* of data obtained

* Only Jens and Lottes (Ref. 88) found that boiling incipience is not affected by gas content.

in boiling experiments where the effect of dissolved gas was evaluated. In several experiments, the effect of dissolved gases was found to diminish in fully developed nucleate boiling (Refs. 24 and 100). This is evident in Fig. 26 at high wall superheat where the data for boiling with dissolved air match that for degassed boiling.

Data obtained to evaluate the effect of dissolved gases on CHF are contradictory; however, in experiments where an effect was noted (Refs. 24, 99-101), the presence of dissolved gases tended to reduce the CHF from that of degassed boiling at the CHF. Fisenko, et al. (Ref. 99) found this reduction to be as high as 23 percent at higher subcoolings. A number of investigators (e.g., Refs. 21, 31, and 102) identified no appreciable effect of dissolved gases on CHF. Kalayda, et al. (Ref. 103) found that for water flow boiling, the gas content had little effect on CHF at pressures below 100 bar, but at higher pressures the CHF was reduced with increased concentrations of nitrogen.

Other experimental results have been obtained with gassed and degassed coolants. Jens and Lottes (Ref. 88) and Buchberg, et al. (Ref. 100) in water flow boiling found that the presence of dissolved nitrogen had a small effect on the pressure loss along the heated wall. Baranenko, et al. (Ref. 104) found for atmospheric pool boiling of water that the bubble diameters are larger when degassed water is used rather than gassed water. The presence of dissolved oxygen can have an additional effect on the heat-transfer process due to oxidation of surfaces in contact with the coolant. A discussion of this oxidation effect is included in the following section.

Surface Aging, Deposits, and Coatings

Surface aging* has been shown in the past to increase the wall superheat in pool boiling (Refs. 11 and 13). Bonilla and Perry (Ref. 73), for instance, found this to be the case on a chrome-plated copper heater with boiling ethanol. Hughes (Ref. 21) found for subcooled flow boiling that surface aging tended to decrease the CHF, and Akhanda and James (Ref. 51) found that aging increased the wall superheat in forced-convection boiling of water.

The effect of surface deposits (or fouling) on cooling heat transfer is much more difficult to ascertain. Epstein (Ref. 105) identified several categories in which fouling (primarily in forced-flow heat exchangers) can occur. These include precipitation (dissolved substance deposits), particulates (suspended particle deposits), chemical reaction (surface is not a

* Nucleate boiling on clean surfaces will not reach a steady-state condition for a sizeable period of time (in some cases, an hour or more) due to several reasons including surface outgassing, chemical reactions, etc. (Ref. 11). This process of reaching steady-state conditions is known as aging.

reactant), and corrosion (surface reactions) among others. Collier (Ref. 7) pointed out that porous deposits on heat-transfer surfaces can influence the CHF both favorably and adversely (see also Westwater, Ref. 11). Bui and Dhir (Ref. 106), found that the presence of an oxide increased the CHF in pool boiling of water. Mel'nikov, et al. (Ref. 107), however, found that CHF decreased nearly 36 percent (at a quality of 0.06) with the presence of a 55-μm (0.002-in.)-thick ferric-oxide scale as compared to a scale-free surface in forced-convection boiling of water.

The effect of surface coatings (primarily used for heat-transfer enhancement) on cooling heat transfer is similar to that of deposits. For example, Zhukov, et al. (Ref. 108) found that the wall superheat and CHF primarily decreased for an aluminum oxide coating, but both increased for an enameled surface in Freon pool boiling. As discussed previously in the wall material effects, Kovalev, et al. (Ref. 77) found that low thermal conductive coatings increased the wall superheat. Collier (Ref. 7) discusses improvements in heat transfer demonstrated by several investigators using thin Teflon® films; however, Sadasivan and Lienhard (Ref. 60) showed that Teflon microcoatings on Nichrome® surfaces reduced the wettability and, hence, reduced CHF values.

Acceleration and Turbulence

High-enthalpy facility components invariably require coolant flow direction and area changes which, in turn, cause flow acceleration/deceleration and turbulence. Merte and Clark (Ref. 109) found for pool boiling of saturated water that increasing the acceleration (g-loading of 1 to 20) decreased the wall temperature and thereby increased the heat-transfer coefficient in both the pure convection and nucleate boiling regions, although the effect of acceleration diminished as the boiling became fully developed. Forced-convection acceleration/turbulence effects have been studied with curved channels and coiled tubes. Gu, et al. (Ref. 110), studying a fluorocarbon liquid flowing over a concave-shaped heater, found that for the same flow velocity and subcooling, both the heat-transfer coefficient and CHF are higher in a curved channel (concave section) than in a straight channel. Gu's team also found that the increase in CHF is significant (40-percent higher than straight channel CHF) at higher flow velocities. Hughes (Ref. 21) found similar results for concave heater surfaces in subcooled Freon 113 flow boiling. Hughes found that the CHF was increased up to 50 percent in concave channels over straight channels (Fig. 27). Hughes' data, however, illustrate the disadvantages of such a curved channel configuration. Should the channel be heated entirely, the convex portion could have a detrimental effect on CHF, as shown in Fig. 27. The CHF for convex surfaces was as much as 22 percent lower than that for a straight channel. Jensen and Bergles (Ref. 95) found low CHF levels in subcooled R-113 flow over inside surfaces (convex portion) of helically coiled tubes, and Bergles (Ref. 8) noted similar results found by other investigators. Winovich and Carlson (Ref. 111) made use of the advantages of a concave surface in the design of an undulating flow channel used in various high-enthalpy facility components. They found that the undulating channel significantly increased (as much as 100 percent) the CHF over that of a straight channel (Fig. 28).

Swirl flow devices and other inserts have been used in the past to induce turbulence, thereby enhancing the cooling heat transfer in applications such as heat exchangers. Kudryavtsev, et al. (Ref. 112), used a twisted tape in tubes with saturated water flow boiling to show that the turbulence generated by density gradients and acceleration from the twisted tapes decreased the rate of nucleate boiling (increased wall superheat). However, Lopina and Bergles (Ref. 113) found little effect of swirl on the wall superheat for subcooled water flow boiling. Ornatskiy, et al. (Ref. 114) demonstrated that induced swirl at the inlet of a concentric annulus increased the CHF by 10 to 60 percent over unswirled flow for low qualities. Gambill, et al. (Ref. 115) found similar results for subcooled water flow boiling. Ornatskiy's team also showed that the effect of swirl on CHF decreases with increasing pressure (Fig. 29), but increases with increasing mass velocity (Fig. 30).

Megerlin, et al. (Ref. 116) augmented the heat transfer in subcooled water flow in tubes by using brush and mesh inserts, showing that heat-transfer coefficients up to nine times the coefficients of empty tubes can be obtained. Megerlin and his team found that the inserts produced very large pressure drops and very large wall superheats. Bergles (Ref. 8) recommends that the inserts are more suitable for use in single-phase flow heat-transfer augmentation. As mentioned previously in the discussion of surface roughness effects, macro-roughness can be used to enhance the heat transfer in forced flow, which probably involves some type of turbulence mechanism.

Heat Flux Distribution

Variations in heat flux distribution, both axial and circumferential, have been shown to have an effect on boiling heat transfer, primarily on the CHF. Stein and Begell (Ref. 117), using subcooled water flowing in internally heated annuli, found no significant effect of an axial heat flux distribution (cosine shape) on the heat-transfer coefficients. Various researchers, however, have found significant effects when CHF is approached. Styrikovich, et al. (Ref. 118), while studying subcooled and saturated water flow in pipes with uniform and linearly varying heat flux, found that the CHF for a linearly increasing heat flux (along the tube length) was nearly twice that of a tube with uniform heat flux for low-quality water at 100 atm (mass velocity of 410 lbm/ft^2 sec) and 40 to 50 percent higher at 180 atm. For these cases, the crisis always occurred at the exit of the heated tube. Styrikovich's team also found that the CHF for a linearly decreasing heat flux occurred near the inlet of the heated section and was larger than that with a uniform heat flux, although data errors prevented the increase from being quantified for the subcooled case. Ornatskiy, et al. (Ref. 119) found similar results for subcooled water flow in annuli with linearly or parabolically increasing heat flux; however, they found quite different results for other nonuniform heat flux applications. Earlier (Ref. 114), Ornatskiy's team had shown that for a cosinusoidal heat flux distribution on annuli with subcooled water flow, the CHF can be as much as 80 percent lower than that of a uniformly heated annulus at similar conditions, and the difference between the nonuniform heating CHF and uniform heating CHF increased with a decrease in pressure and mass flow rate. In addition, data presented in Ref. 119 show that, for subcooled water flow in annuli, the CHF with sinusoidal, exponential, or linearly/parabolically decreasing heat flux is generally lower than the CHF with uniform heat flux. Similar results were presented by Zenkevich, et al. (Ref. 90) for subcooled and saturated water flowing in tubes with uniform and cosinusoidal heat flux distribution, although at higher pressure and subcooled conditions, the difference appears to diminish or even reverse (Fig. 31). They point out that for complex nonuniform heat flux applications, the location of the burnout point along the length of tube or annulus is difficult to predict. Swenson, et al. (Ref. 120) presented some interesting results concerning saturated water flow in tubes with nonuniform heating along their length. They showed that the effect of nonuniform heating on DNB (CHF) was similar to that of Ornatskiy's team (CHF for nonuniform heating is less than CHF for uniform heating); however, as the region over which the nonuniform heating occurred became very small (i.e., a spike heat flux), the CHF for the nonuniform heating case approached and surpassed that for a uniformly heated tube.

Aladyev, et al. (Ref. 45) demonstrated that nonuniform heating around the perimeter of a tube with subcooled or saturated water flow can affect the level of CHF. They showed that the CHF generally occurs where the heat flux is the highest and increases with increasing unevenness of the heating (characterized by the ratio of the maximum to the average heat flux). It was also shown that the effect of nonuniform heating around the perimeter decreased with increasing diameter. Leontiev, et al. (Ref. 121) found, for subcooled water flowing in horizontal tubes with nonuniform circumferential heating (the ratio of maximum heat flux to average heat flux was 1.5), that the local values of CHF were somewhat higher than those of uniformly heated tubes. However, when they averaged the heat flux at the station where the critical point occurred, the averaged CHF for the nonuniform heating case was much lower than that for the uniform heating case at subcooled conditions, and the two coincided at high-quality conditions.

Transient Operation

The effects of transient power (heat flux), flow rate, and pressure operation on heat transfer have been studied in previous experiments primarily associated with petrochemical and nuclear power plant safety (e.g., Ref. 122). A major portion of these experiments involved the study of transient heat flux effects. Ragheb, et al. (Ref. 123), showed that the wall superheat was larger for transient boiling (found by quenching a high thermal capacity tube with subcooled, low-pressure water flow) than steady-state boiling at comparable conditions (Fig. 32). Cheng, et al. (Ref. 124) had earlier found little difference in wall superheat for transient or steady-state heating with the same apparatus, although a limitation in the steady-state heating capability prevented data comparison at heat flux levels approaching the CHF. Kataoka, et al. (Ref. 125) found an effect on wall superheat similar to the one found by Ragheb's team for an exponentially increasing heat input to a platinum wire in subcooled forced water flow.

Several interesting results have been found during studies of the effect of transient heat flux on CHF. Borishanskiy and Fokin (Ref. 64) and later Tolubinskiy, et al. (Ref. 126) found that the CHF was equal for transient and steady-state heating in pool and forced-convection boiling of water with and without subcooling. However, when organic coolants* were used, it was found that the CHF for steady-state heating was much larger than the CHF for transient heating (Tolubinskiy's team found this difference to be as large as a factor of 3). Tolubinskiy's team theorized that the transient operation is viscosity-dependent, and pointed out that the organics have higher viscosities than that of water, thereby contributing to the difference in CHF for the transient and steady-state heating of organic fluids. Roemer (Ref. 127) found no effect of transient heating (time constant of the applied power was approximately 0.1 sec) on CHF for water flow normal to cylindrical test elements as long as the test surfaces were aged. This result was found to be independent of the test element wall thickness and the speed of the power transient. The CHF for unaged specimens was as much as 25 percent lower than the CHF at comparable steady-state conditions. Other researchers have found results conflicting with those observed by Borishanskiy's team and Tolubinskiy's team for water. As seen in Fig. 32, the data of Ragheb, et al. (Ref. 123) show the CHF for transient conditions to be higher than that for steady-state conditions. Kataoka, et al. (Ref. 125) also found the CHF for transient heating to be greater than that for steady-state heating, and the difference between the two CHF varied approximately as the -0.6 power of the period of the exponentially increasing heat flux (Fig. 33).

Fewer experiments have been performed to evaluate the effects of transient flow rate or pressure operation on boiling heat transfer. Celata, et al. (Ref. 128), using subcooled Refrigerant-12 as a coolant, found that the difference between the CHF for a flow rate transient and the CHF for a steady-state flow rate became significant when the time to reach half of the initial flow became less than approximately twice the flow rate transit time. The CHF for the transient flow was typically less than the comparable steady-state CHF in these cases. Celata, et al. (Ref. 129) later evaluated the combined effect of transient flow rate and thermal power on CHF using flowing subcooled R-12. His team found that the CHF for the transient

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* Borishanskiy's team used ethyl alcohol; Tolubinskiy's team used ethanol and acetone.

cases was approximately equal to that for steady-state conditions when the time to reach half flow was greater than the transit time. When the time to reach half flow dropped below the transit time, the CHF for the transient case became greater than the corresponding steady-state CHF. These trends appeared to be insensitive to the level of pressure or mass flux at which the runs were made.

Weisman, et al. (Ref. 130), using depressurization of saturated water to produce boiling in a nonflowing apparatus, found that the wall superheat for the pressure transients was significantly higher (as high as a factor of two) than a slow pressure change, although the data are probably applicable only near boiling incipience. Celata, et al. (Ref. 131) used flowing, subcooled R-12 to demonstrate that pressure transients tended to reduce the CHF (higher depressurization rates led to more marked crisis conditions) from that exhibited during comparable steady-state conditions. Finally, Celata, et al. (Ref. 122) evaluated the combined effect of transient pressure, power, and/or flow rate on CHF, although no steady-state data were obtained for comparison. Celata's team did show the inadequacy of using the available steady-state CHF correlations in predicting transient power/flow rate/pressure situations. The data acquired by Celata's team for one-, two-, and three-parameter transients were recently tabulated in Ref. 132.

Flow Instabilities

Thermal-hydraulic flow instabilities are directly related to the flow system design, and can seriously affect the value of CHF and the integrity of the flow system (Ref. 17). Podowski (Ref. 133) identified several instabilities that can appear in two-phase flow and pointed out that these instabilities can be compounded in some cases. The classes of instabilities include excursive (Ledinegg), flow regime relaxation, nucleation, density-wave oscillation, pressure drop oscillation, acoustic, and condensation-induced instabilities. Numerous authors (Refs. 17, 133-136) have provided excellent reviews of these instabilities and their effects; therefore, a rigorous treatment is excluded here. In general, flow instabilities have a detrimental effect on boiling heat transfer, primarily a reduction in the CHF (Ref. 14).

According to Podowski (Ref. 133), one of the most predominant of the instabilities is that classified under density-wave oscillation. It includes flow loop, parallel-channel, and channel-to-channel instabilities. Flow loops consisting of sections with liquid flow combined with sections having liquid/vapor flow will have areas with different speeds of propagation of perturbations. In such a system, a perturbation can induce velocity and pressure oscillations which may diverge or reach a self-sustained periodic mode. In the latter case, the oscillations can exceed the thermal limits of the system, thereby reaching a premature CHF. Premature CHF has also been demonstrated in parallel channels (see Veziroglu and Lee, Ref. 137) that share a common inlet manifold or have a bypass channel. In configurations where the channels have similar operating conditions, the phase of oscillations may be opposite, depending on the number of channels (Ref. 133). Such oscillations are called channel-to-channel instabilities. In many cases, the manifolded cooling approach is used in high-enthalpy arc facilities and may be prone to these types of instabilities when two-phase flow is present. Concepts for future arc facility nozzle liners include thin walls with structural webbing or ribs on the cooling channel surface. Such designs may introduce channel-to-channel instabilities between the cooling passages on the nozzle liners.

Density-wave oscillations frequently combine with other phenomena such as pressure-drop oscillations to create additional instabilities. For instance, Lowdermilk, et al. (Ref. 101) and later Aladyev, et al. (Ref. 45) showed that pulsations in flowing subcooled and saturated water caused by an upstream compressible volume reduced the CHF (from that of a stable system with no pulsations) as much as 80 percent. Lowdermilk's team found that the flow instability could be removed by throttling the flow just upstream of the test section (Fig. 34). In addition, Gambill, et al. (Ref. 115) suggested that stability is improved as the test section pressure drop becomes a smaller fraction of the total system pressure drop, and restricted flow upstream of the test section for most of their tests.* It appears that the stability of a cooling apparatus which involves flow boiling can be improved by avoiding compressible volumes in the flow loop (or throttling the flow between the compressible volume and the test section) and forcing the test section pressure drop to be a small portion of the total pressure drop of the system. However, as pointed out by Boyd (Ref. 17), in cases where a compressible volume is in the test section, or in very long test sections, no amount of throttling can eliminate or reduce the instability.

Heat Application/Orientation

Although not typically associated with an actual cooling configuration, the method of heat application when resistive (Joule) heating is used in an experimental simulation can be important. Ellion (Ref. 138) was one of the first to note an effect of alternating current (a-c) heating on an experimental boiling apparatus. Ellion found that a 60-cycle/sec a-c power supply caused a 120-cycle/sec growth and collapse of the bubbles in flowing subcooled water in an annulus. The problem was rectified by switching to direct current (d-c) power with the amplitude of the voltage ripple less than 1 percent over the full operational range. Gambill, et al. (Ref. 115) suspected a similar problem when they used a-c power to electrically heat a tube with internally flowing subcooled water. Gambill's team calculated that the a-c heating reduced the CHF as much as 16 percent over that for d-c heating. Tippets (Ref. 28), who used a-c heating of stainless steel ribbons to study subcooled and saturated water flow boiling,

* Gambill's team restricted the ratio of test section pressure drop to the total system pressure drop to values between 0.045 and 0.40.

found that the CHF for 0.006-in.-thick heater ribbons averaged 20 percent lower than those for 0.010-in.-thick ribbons, and in a few cases, the difference was substantially larger. Tippets theorized that the reduction was due in part to the lesser thermal time constant of the thinner ribbons coupled with the a-c heating. Although Sadasivan and Lienhard (Ref. 60) showed a definite difference between a-c and d-c heating on bubble character, they found little effect on CHF.

Leung, et al. (Ref. 44) found little effect (less than 6-percent difference) of direct and indirect heating of the interior wall of an annulus with flowing water at subcooled inlet conditions. As mentioned previously, Collier (Ref. 7) points out that the outer wall in a concentric annulus system has little or no effect on the CHF for the inner surface. However, Tong (Ref. 10) suggests that the presence of an unheated wall in the proximity of a critical point can adversely affect the cooling effectiveness (termed the "unheated wall effect"). Tolubinsky, et al. (Ref. 139) found, for subcooled and saturated water flowing in annuli, that higher CHF (as much as 30 percent) existed on a heated inner wall than on a heated outer wall at comparable conditions. Yucel and Kakac (Ref. 140) found that the heater orientation had an effect on the CHF in a rectangular channel with flowing subcooled water. Their data indicated that higher CHF is achieved with the heated surface (along one side of the rectangular channel) facing up rather than facing down. However, they attributed the difference not to the presence of an unheated wall, but to buoyancy forces moving the vapor away from the lower surface, resulting in a lower surface temperature and higher CHF (as opposed to a higher surface temperature and lower CHF on the downward facing heater).

Miscellaneous Effects

Several other parameters have been shown to have an effect on the cooling process, albeit only a small number of experiments have been performed that demonstrate their effect. Tong (Ref. 10) suggested that the local enthalpy can impair the CHF, primarily when there is significant vapor voidage near the heated wall. Such conditions could exist for subcooled* or saturated flow, even at high velocity. The local enthalpy near the wall, even at upstream locations, can be excessively high and can cause burnout when significant vapor exists in the bubble layer near the wall.

Johnson (Ref. 141) pointed out that the presence of an electric field has some effect in the pure convection region, little effect in the nucleate boiling region, and a significant effect

* Fiori and Bergles (Ref. 20) found, contrary to prior beliefs, that large vapor voidages can exist even in subcooled flow. They noted that instantaneous void fractions larger than 50 percent were possible at high heat flux (3.1×10^6 to 17.3×10^6 W/m^2), high exit subcooling (28° to 56°C), and low pressure (less than 6.2 bar).

at CHF and beyond. Markels and Durfee (Ref. 142) applied a 3,000-v a-c voltage across the gap of an annulus with the inner wall heated and flowing subcooled water. They found that the CHF was augmented as much as 40 percent with the applied electric field as opposed to no field present. No effect of the applied voltage was found where flow instability burnouts occurred.

Another heat flux augmentation method which has been demonstrated involves the use of flow and ultrasonic vibration. Although the use of vibration has not resulted in an increase in CHF for forced-flow configurations (Refs. 7 and 143), an enhancement of the heat transfer (increase in the heat-transfer coefficient) in the single-phase forced-convection region and boiling onset has been demonstrated (see Bergles, Ref. 143).

Boyd (Ref. 1) identified two other heat flux enhancement techniques which should be included here for completeness. As Boyd noted, Inoue and Bankoff (Ref. 144) found that the destabilization of film boiling (initially subcooled) by the introduction of a pressure wave increased the transient heat transfer up to 20 times the steady-state values. Boyd also noted that Weede and Dhir (Ref. 145) showed that the CHF can be enhanced and its location controlled using tangential injection of a fluid into the mainstream. This technique probably takes advantage of the induced swirl and turbulence whose benefits were discussed previously.

3.2 SUMMARY OF PARAMETRIC TRENDS AND IMPLICATIONS

As suggested previously, velocity and subcooling have a primary contribution on cooling effectiveness, and it appears that a highly subcooled, forced-convection cooling system is most attractive for high-enthalpy arc facility component cooling. Figure 35 summarizes the effect of a majority of the parameters and enhancement techniques previously discussed for a subcooled, forced-convection cooling environment. Those parameters and techniques that tend to decrease heat transfer or increase wall temperature in the pure convection (nonboiling) and nucleate boiling regions, respectively, are not necessarily detrimental to the cooling process unless the wall temperature increase approaches the melt temperature of the material or contributes to plasticity to the point of structural failure. Probably the most important contributions to cooling effectiveness and, hence, component survival, are those parameters and enhancement techniques which tend to increase the CHF. A number of parameters, such as pressure, can have either a beneficial or a detrimental effect on CHF and must be optimized for the conditions of interest. However, system limitations and configuration requirements may limit the degree of optimization that can be achieved.

The implications of requiring a subcooled, forced-convection cooling configuration are numerous and are not generally conducive to experimental evaluation. As previously discussed, high velocity, high subcooling, and high pressure tend to decrease bubble diameter and lifespan,

where

μ_b/μ_w is between 0 and 40

Re_D is between 10^4 and 5×10^6

Pr is between 0.5 and 200 for 6-percent accuracy

Pr is between 200 and 2,000 for 10-percent accuracy

$n = 0.11$ for $T_w > T_b$

$n = 0.25$ for $T_w < T_b$

$n = 0$ for gases

and where

$f = 1/(1.82 \log_{10} Re_D - 1.64)^2$ for smooth pipes

or from Moody chart for smooth or rough pipes.

Sleicher and Rouse (Ref. 152) recommended a simpler expression with comparable accuracy for tube flow,

$$Nu_D = 5 + 0.015 (Re_D)^a (Pr)^b \tag{6}$$

where

$a = 0.88 - 0.24 / (4 + Pr)$

$b = (1/3) + 0.5\, e^{(-0.6\, Pr)}$

and

Pr is between 0.1 and 10^5

Re_D is between 10^4 and 10^6

The properties in Re_D are evaluated at the local film temperature, $T_f = (T_b + T_w)/2$, and the properties in Pr are evaluated at the local wall temperature.

Kays and Leung (Ref. 153) solved the turbulent-flow energy equation for constant heat flux in annuli over wide ranges of Reynolds number, Prandtl number, and annulus radius ratio. Table 1 in Ref. 153 or Tables 13-3 to 13-5 in Ref. 5 present the results of their computations.

For fully developed laminar flow in a circular tube with uniform heat flux, the Nusselt number and, hence, the heat-transfer coefficient is constant, independent of Reynolds number and Prandtl number,

$$Nu_D = 4.36 \qquad \text{constant heat flux, Pr} > 0.6 \tag{7}$$

Treatment of heat transfer in the entry region where velocity and temperature profiles are not fully developed is discussed in Chapter 8 of Ref. 154.

Pressure drop is also of interest in a cooling process, and the pressure gradient has even been shown to have an effect on local heat transfer (see Chapter 6 of Ref. 148). Solution of the equations of motion (using semiempirical shear correlations for the turbulent case where random fluctuations are important) provides a means of determining the pressure drop in a flow cooling configuration as outlined by White (Ref. 155). Friction factors as a function of Reynolds number for flow in smooth or rough circular tubes, annuli, or noncircular ducts are provided by the Moody chart or expressions such as those presented in Chapter 6 of Ref. 155.

4.2 TRANSITION TO FLOW BOILING, FLOW BOILING, AND CHF: THEORETICAL MODELS

The theoretical determination of heat transfer, pressure drop, and other flow characteristics once a cooling process has transitioned into boiling is one of the greatest challenges facing researchers in two-phase flow. Three currently accepted two-phase flow or "mixture" models have been developed based on the more dominant flow regimes in two-phase flow. The *drift flux* model, which is based on a molecular diffusion model, allows for motion of one phase relative to the other, and the two are coupled through the use of correlations accounting for interaction effects. The correlations are difficult to obtain because they require a knowledge of the average effects of one phase moving relative to the other (see, for example, Ref. 156). Another theoretical model, the *two-fluid* model based on the application of the Navier-Stokes equations, treats the phases separately and requires matching conditions at both solid boundaries and liquid-vapor interfaces. Difficulty is encountered in obtaining local and instantaneous mass, momentum, and energy balance equations at the interfaces (see, for example, Ref. 157). The drift flux model is typically more applicable to flows where the one phase is dispersed throughout the other. The two-fluid model is generally used for flows where the two phases are separated into layers, but has been accepted for use with dispersed flows. If the motion of the two phases can be assumed to be equivalent (i.e., no-slip condition), a simplified model, the *homogeneous* model that treats the mixture as a single-phase fluid, may be used. The transport properties required for the model are difficult to determine due to the lack of existing relationships. Although extensive progress in theoretical modeling has been achieved in recent years, its application has been limited because of the difficulties described above. A more complete treatment of these models, especially the two-fluid model, is included in Jones, Ref. 158, and Drew, Ref. 159.

4.3 TRANSITION TO FLOW BOILING: CORRELATIONAL APPROACHES

Early studies in flow boiling suggested that boiling incipience occurred at the intersection of the pure convection and fully developed nucleate boiling curves, although experimental data indicated a transition region (McAdams, et al., Ref. 24). Several investigators later derived heat-transfer expressions for the transition from pure convection to fully developed nucleate boiling (termed partial nucleate boiling). Kutateladze (Ref. 160) and, later, Forster and Greif (Ref. 57) suggested correlations based on the pure convection heat flux at the incipience of boiling and the corresponding value of heat flux in pool boiling. Bergles and Rohsenow (Ref. 161), however, later showed that the transition region between pure convection and nucleate boiling cannot be based on results from saturated pool boiling, and recommended the following relation based on an extension of the fully developed nucleate boiling curve,

$$q''/q''_{conv} = [1 + \{(q''_B/q''_{conv})(1 - q''_{Bi}/q''_B)\}^2]^{1/2} \tag{8}$$

where q''_B is calculated from a fully developed nucleate boiling correlation at various wall temperatures, and q''_{Bi} is determined from an extension of the fully developed nucleate boiling curve to the temperature corresponding to the boiling incipient point (see Fig. 37). In addition, Bergles and Rohsenow developed a relation for heat flux at subcooled boiling incipience by solving a bubble growth equation graphically to obtain

$$q''_{LB} = 15.60\, p^{1.156} (T_w - T_{sat})^{2.3/p^{0.0234}} \tag{9}$$

4.4 FLOW BOILING: CORRELATIONAL APPROACHES

The team of McAdams, et al. (Ref. 24) was one of the earliest boiling research teams to propose an empirical correlation for fully developed nucleate boiling in subcooled forced flow. They found for flow in an annulus (heated stainless steel inner wall) that the heat flux during boiling was independent of coolant velocity (1 to 36 ft/sec), degree of inlet subcooling (20° to 150°F), pressure (30 to 90 psia), and equivalent diameter (0.17 to 0.48 in.). They correlated the heat flux data with the expression,

$$q'' = C' \Delta T_{sat}^{3.86} \tag{10}$$

where the value of C′ decreased from 0.19 to 0.074 as the concentration of dissolved gas decreased from 0.30 to 0.06 ml of air at standard conditions per liter of water. Jens and Lottes (Ref. 88) analyzed high-pressure, high heat flux, flow boiling data obtained by researchers at UCLA, Purdue, and MIT, and derived the following empirical correlation,

$$q'' = (0.527\, e^{p/900}\, \Delta T_{sat})^4 \tag{11}$$

where p is in pounds per square foot. Jens and Lottes found that the wall temperature during boiling was independent of the flow velocity, gas content, and degree of subcooling and was dependent only on heat flux and water pressure. All of the data were obtained for flow inside round or square tubes made of stainless steel or nickel.

Gilmour (Ref. 162) derived a correlation based on dimensionless groups rather than temperature difference:

$$(h/c_\ell\, G)\, (c_\ell\, \mu_\ell/k_\ell)^{0.6}\, (\varrho_\ell \sigma/p^2)^{0.425} = 0.001/(DG/\mu_\ell)^{0.3} \tag{12}$$

where

$$G = (m_v/A)\, (\varrho_\ell/\varrho_v)$$

and where p is in pounds per square foot. Although Gilmour substantiated the correlation with data from various experiments, including several in which water was used, it was noted that surface material had a significant effect that was not accounted for in the correlation and that data from experiments involving boiling from wires did not agree well with the correlation.

Later, Levy (Ref. 163) derived a semiempirical correlation based on the Forster and Zuber (Ref. 164) bubble growth model and applicable to nucleate boiling with flow as well as pool boiling, arguing that the fluid velocity normally used exerts a minor effect in comparison with the local turbulence produced by the bubble motions. For subcooled liquids, Levy proposed the correlation,

$$q'' = [k_\ell c_\ell \varrho_\ell^2/\sigma T_s\, (\varrho_\ell - \varrho_v)]\, [1/B_L]\, [\{i_{fg} + c_\ell\, (T_s - T_b)\} / i_{fg}]\, (\Delta T_{sat})^3 \tag{13}$$

where T_s and T_b are in °R, σ in Btu/ft^2, and B_L is an empirical constant which is a function of $\varrho_v\, i_{fg}$ as shown in Fig. 38. Levy found that the above correlation agreed reasonably well with data from McAdams, et al., and the UCLA data used by Jens and Lottes discussed previously.

About the same time Levy derived his correlation, Forster and Greif (Ref. 57) theorized that the primary boiling heat-transfer mechanism is the exchange between the vapor and liquid, and that microconvection in the liquid sublayer beneath the bubbles, bubbles acting as artificial roughness, and latent heat transport by the bubbles contribute very little. Based on this argument, they used a Reynold's analogy to derive the following expression for general boiling of various liquids including water:

$$q'' = K_{sf}\,[\alpha c_\ell \varrho_\ell T_s / J i_{fg} \varrho_v \sigma^{1/2}]\,[c_\ell T_s \alpha^{1/2} / J\,(i_{fg} \varrho_v)^2]^{1/4}\,[\varrho_\ell/\mu_\ell]^{5/8}\,[\mu_\ell c_\ell / k_\ell]^{1/3}\,\Delta p^2 \qquad (14)$$

where Δp is the pressure difference in pounds per square foot corresponding to the superheat, ΔT_{sat}. Reasonable agreement was achieved when the correlation was compared to experimental data at pressures up to 50 atm with K_{sf} equal to 0.0012. In the discussion in Ref. 57, the authors noted that the constant does change for other data.

Shah (Ref. 165) presented a correlation for the generalized prediction of heat transfer during subcooled boiling in annuli based on earlier work involving tubes (Ref. 166). His correlation is

$$q'' = h_\ell\,(T_w - T_b) + h_\ell\,(\psi_o - 1)\,(T_w - T_{sat}) \qquad (15)$$

where

$\psi_o = 1 + 46\, B_o^{0.5}$ for $B_o < 0.3 \times 10^{-4}$

$\psi_o = 230\, B_o^{0.5}$ for $B_o > 0.3 \times 10^{-4}$

$B_o = q''/G i_{fg}$

h_ℓ = nonboiling heat-transfer coefficient

Shah found that the expression correlated 450 representative data points with a mean deviation of 8.9 percent. The range of test conditions was: reduced pressure from 0.009 to 0.89, inlet subcooling from 2° to 109°C (3.6° to 196°F), mass flux from 278 to 7,774 kg/m^2 sec (57 to 1,592 lbm/ft^2 sec), inner tube diameter from 4.5 to 42.2 mm (0.177 to 1.66 in.), and annular gap from 1 to 6.6 mm (0.039 to 0.26 in.). Various fluids, including water, were evaluated and heating was from the inner, outer, or both walls of the annuli.

Probably the most widely accepted and verified flow boiling correlation is based on Rohsenow's (Ref. 167) pool boiling correlation (a Nusselt number derived from the fluid Prandtl number and a bubble Reynolds number) and proposed by Rohsenow (Ref. 168) in

1952. The heat-transfer rate associated with forced-convection boiling was found to be accurately predicted by addition (or superposition) of the pool boiling effect with the pure forced-convection effect discussed above, i.e.,

$$q'' = q''_{\text{forced convection}} + q''_{\text{pool boiling}} \tag{16}$$

where $q''_{\text{forced convection}}$ is determined from the expressions presented in the previous discussion of pure forced-convection correlations and $q''_{\text{pool boiling}}$ is determined from

$$c_\ell\,(T_w - T_{sat}) \,/\, i_{fg}Pr^s = C_{sf}\,[\{q''/\mu_\ell i_{fg}\}\,\{g_c\,\sigma/g\,(\varrho_\ell - \varrho_v)\}^{1/2}]^r \tag{17}$$

where the properties are evaluated at T_{sat} (except as noted), r equals 0.33, s equals 1 for water cooling and 1.7 for other fluids, and C_{sf} equals 0.006 for nickel and brass surfaces and 0.013 for copper and platinum surfaces.* Vachon, et al. (Ref. 169) later presented new values of r and C_{sf} which account for surface preparation technique as well as liquid-surface combination. However, their analysis applies only to the linear portion of the nucleate boiling curve, i.e., that portion which excludes partial nucleate boiling and approaching CHF. In addition, the effects of subcooling and pressure have not been fully investigated and may affect the constants presented by Vachon's team. Tong (Ref. 10) pointed out that the superposition approach may be questionable in light of the experiments of Bergles and Rohsenow (Ref. 161) which demonstrated the differences in the fluid mechanics of flow boiling and pool boiling. Bergles and Rohsenow recommended a less restrictive construction approach for predicting forced-convection boiling when the net vapor generation is relatively small, which was presented previously in Eq. (8). Bjorge, et al. (Ref. 170) and Stephan and Auracher (Ref. 171) later presented variations of the superposition approach. For additional approaches, Guglielmini, et al. (Ref. 172) provided a survey of flow boiling correlations as of 1980, including boiling incipience and partial nucleate boiling expressions.

Large vapor void fractions have been observed in low-pressure, subcooled flow, possibly greater than 20 percent on a time-averaged basis and exceeding 50 percent on a local instantaneous basis (Ref. 20). Such void fractions contribute to the accelerative and static head components of the pressure gradient along the heated channel. Flow boiling experiments where pressure drop characteristics were evaluated (e.g., Refs. 22 and 173) generally have shown that as heat flux increases, the pressure drop decreases until boiling begins, after which the pressure drop increases (Fig. 39). Boyd (Ref. 174), however, found pressure drops for high-pressure, moderately subcooled water boiling to be slightly less than single-phase pressure drops. Boyd noted that CHF occurred before substantial pressure drops occurred. Bergles

* The Rohsenow correlation is valid only for clean, relatively smooth surfaces. Rohsenow (Ref. 167) found that s varied erratically between 0.8 and 2.0 when the surface was dirty.

and Dormer (Ref. 173), who performed experiments in subcooled water flow boiling at pressure below 100 psia, noted that little success had been reported in the general correlation of pressure drop, and presented graphical correlations for the pressure drops encountered in their experiments. About the same time, Staub and Walmet (Ref. 175) identified the two regions before and after the point of significant vapor generation (SNVG) where the effects of void fraction on pressure drop are different in subcooled flow boling. Recently, Hoffman and Kline (Ref. 176) and Lu and Jia (Ref. 177) have presented correlations for the prediction of void fraction and the subsequent gravity, acceleration, and friction pressure drop before and after SNVG.

4.5 CHF CORRELATIONAL APPROACHES

No theory of subcooled burnout has yet been created for flow boiling (Ref. 33). Complexity of the boiling mechanisms at the critical point, their unsteady nature, and the small scale of the heat-transfer processes have hindered the development of theoretical models and understanding of the phenomena from experimental studies. Moreover, the cooling flow structure and, hence, the heat-transfer processes vary depending on the heating condition, fluid properties, and flow/surface geometries as indicated in the previous discussion of parametric effects. This has led to an excessive number of correlations derived for the prediciton of CHF in flow boiling. Most of the CHF correlations, not unlike those for fully developed nucleate boiling, were developed either by the analytical (mechanism-based) approach, dimensional analysis/similitude-based approach, or optimization of test data (empirical-based).

4.5.1 Mechanism-Based CHF Correlations

In an attempt to derive analytical relationships among the macroscopic heat-transfer and flow-field characteristics in a boiling environment, several models describing the mechanisms on a microscopic scale have been proposed. The mechanisms appear to vary with the degree of subcooling, and several current approaches divide the subcooled boiling regions into three areas: (1) high subcooling and mass velocity, (2) moderate subcooling, and (3) low subcooling, velocity, and pressure (Ref. 8). At high subcooling and mass velocity, it has been proposed that local wall overheat occurs when high heat flux leads to dryout and a resulting hot spot beneath a bubble; however, there is little experimental support, and no correlations have been developed based on this mechanism (Ref. 8). At moderate subcoolings, it has been proposed that the liquid flow stagnates near the heated wall when the boundary layer separates. At high heat flux, the stagnant liquid evaporates, resulting in vapor blanketing near the wall and subsequent burnout. Tong (Ref. 178) presented a correlation based on this mechanism which agreed reasonably well with data from subcooled, high-pressure water flow in tubes, annuli, and rod bundles. Purcupile and Gouse (Ref. 179) presented a similarly based correlation in nondimensional form. Other studies associated with this proposed mechanism include the work of Kutateladze and Leont'ev (Ref. 180) and Hancox and Nicoll (Ref. 181). At low subcooling, velocity, and pressure, Fiori and Bergles (Ref. 20) suggested that dry-out of the microlayer follows when inadequate quenching of the surface by adjacent liquid occurs after the passage of a bubble. Subsequent burnout occurs when high surface temperatures prevent rewetting of the surface. Because of difficulties in defining the flow structure, no correlation could be obtained by the authors. Lee and Mudawwar (Refs. 182 and 183) proposed a similar mechanism by suggesting that CHF occurs as a result of the Helmholtz instability at the microlayer-vapor interface, leading to dry-out of the microlayer. They demonstrated that their correlation agreed with numerous subcooled (over a wide range of inlet subcoolings) water data points with a mean deviation of approximately 11 percent. Katto (Refs. 184 and 185) presented a correlation based on this same sublayer dry-out mechanism and suggested that CHF of subcooled flow boiling has strong similarities to pool boiling CHF. Earlier, Katto first presented generalized correlations for CHF in subcooled and saturated water flow in tubes (Refs. 186 and 187) and annuli (Ref. 188) based on the existence of four characteristic regimes of CHF and the following dimensionless groups:

$$q''_{CHF} \,/\, \{G i_{fg} \,[1 + K\,(\Delta i_i / i_{fg})]\} = f\,(\varrho_v/\varrho_\ell, \sigma\varrho_\ell/G^2L, L/D) \tag{18}$$

where K is determined analytically to account for inlet subcooling. A reasonable correspondence between the correlations and experimental data was found; however, later studies (e.g., Ref. 189) identified problems with applying the correlations to the various regimes. In addition, difficulty was encountered in deriving an expression for CHF at very high mass velocity and pressure conditions. Nishikawa, et al. (Ref. 190) later correlated high-pressure tube data at various Freon mass flows with Katto's proposed dimensionless groups using the boiling length as the characteristic length. Their proposed expressions correlated approximately 150 data points to within ± 10 percent.

Other proposed mechanisms of general subcooled flow-boiling CHF include liquid flow blockage models and vapor removal limit/bubble crowding models (Ref. 184). The liquid flow blockage models assume that the flow of liquid normal to the heated surface is blocked by the flow of vapor along the surface. The mechanisms proposed by Bergel'son (Ref. 191) and Smogalev (Ref. 192) involve some form of liquid flow blockage. Griffith (Ref. 193) presented a dimensionless correlation based on a critical packing of bubbles on the surface. Griffith proposed that a saturated pool boiling bubble growth term be multiplied by a correction factor which included velocity and subcooling effects. Approximately 94 percent of his data were correlated by the expression to within ± 33 percent. Hughes (Ref. 21) suggested that CHF occurred when a maximum number of nucleation sites available for a surface was exceeded (i.e., bubble crowding). For Freon flow over a straight surface, he proposed the correlation:

$$q''_{CHF} = [2d_s\varrho_v\,(c_\ell\Delta T_{sub} + 2i_{fg})] \,/\, 3K\,\{4/9\pi\alpha\,(\varrho_v\Delta T_{bulk}d_s/\varrho_\ell\Delta T_{sat})^2 + \pi\alpha/3\,(\varrho_v i_{fg}d_s/k_\ell\Delta T_{sat})^2 + (1/B)\,\ln\,[(U_o + Bd_s)\,/\,U_o]\} \quad (19)$$

where

$$d_s = 6.12\ W\ D_h \,/\, Re^{7/8} \qquad (W = 30 \text{ in Ref. 21})$$

$$B = (12/11)\,(C_D\mu_\ell/\varrho_\ell d^2)$$

$$U_o = (3/4\pi\alpha)\,(k_\ell^2\ \Delta T_{sat}{}^2 \,/\, \varrho_v^2\ i_{fg}{}^2 d)$$

and q''_{CHF} is in Btu/in.^{2} sec, ϱ is in lbm/in.^{3}, k is in Btu/in. sec °F, μ is in lbm/in. sec, D_h and d are in in., and α is in in.^{2}/sec. The term K was determined experimentally as

$$K = \beta_{fs}\ V_m{}^{1/2}/\Delta T_{sub} \qquad (\beta_{fs} \text{ is approximately equal to 7.94})$$

Although not given by Hughes, the drag coefficient for a bubble is approximated by

$$C_{D_{sphere}} \approx \frac{24}{Re} + \frac{6}{1 + \sqrt{Re}} + 0.4 \qquad (\text{see Ref. 148, Chap. 3})$$

and, for a straight test section, the bubble diameter for the preceding equations is approximated by

$$d = d_s$$

Hughes also presented somewhat cumbersome algebraic expressions for CHF on convex and concave surfaces (see Ref. 21). Hebel, et al. (Ref. 194) similarly suggested that CHF occurs when a limiting vapor removal rate is reached. Other proposed mechanisms included in this group are those suggested by Yagov and Puzin (Ref. 195) and later, Weisman and Ileslamlou (Ref. 196). Weisman and Ileslamlou (see also Yang and Weisman, Ref. 197), by extending the earlier work reported in Ref. 198 based on limited turbulent interchange at the outer edge of a bubbly layer near a heated surface, proposed the following CHF correlation for use in highly subcooled flow:

$$q''_{CHF} = G\psi i_b\ [i_\ell\,(1 - x_2) + i_v\,(x_2) - \bar{i}] \qquad (20)$$

where

$$\psi = [1/(2)^{1/2}\pi] \exp[-1/2 (v_{11}/\sigma_v')^2] - 1/2 (v_{11}/\sigma_v') \operatorname{erfc}(v_{11} / (2)^{1/2} \sigma_v')$$

$$i_b = 0.462 (K)^{0.6} (Re)^{-0.1} (D_b/D)^{0.6} [1 + a (\varrho_\ell - \varrho_v) / \varrho_v]$$

$\bar{i}$ is the average fluid enthalpy

x_2 is the average quality in the boundary layer corresponding to a void fraction of 0.82

$K = 2.4$ (experimentally determined)

$$D_b = [0.015 (\sigma D_h/\tau_w)^{1/2}] \{1 + \underbrace{0.1 (g/g_c) [(\varrho_\ell - \varrho_v) / \tau_w] D_h}_{\text{buoyancy correction term} \approx 1 \text{ for high mass velocity}}\}^{-1/2}$$

$$v_{11} = q'' i_{fg} (x_2) / [\{i_\ell (1 - x_2) + i_v (x_2) - \bar{i}] (\varrho_v i_{fg})\}$$

$$\sigma_v' = (G/\varrho_\ell) i_b$$

$$a = 0.87 (1.36 - V_{SL}) + 0.135 \qquad 0.5 < V_{SL} < 1.36 \text{ m/sec}$$

$$a = 0.135 \qquad G < 9.7 \times 10^6 \text{ kg/m}^2 \text{ hr}, V_{SL} \geq 1.36 \text{ m/sec}$$

$$a = 0.135 (G/9.7 \times 10^6)^{-0.3} \qquad G > 9.7 \times 10^6 \text{ kg/m}^2\text{hr}$$

where V_{SL} is the superficial velocity of two-phase mixture in m/sec. The correlation requires iteration on q''_{CHF} since v_{11} is dependent on q''. Andreyev, et al. (Ref. 199) suggested that CHF that occurs in the transition from bubbly to dispersed-annular flow can be based on a transition void fraction and recommended an equation for this specific void fraction that agreed reasonably well with experimental data for subcooled water flow boiling. Povarnin (Ref. 200) assumed a concept of corresponding states and derived a correlation that agreed reasonably well with subcooled CHF data for various fluids including water. No further work has been performed in recent years to substantiate Povarnin's approach. Bergles (Ref. 8) identified several other proposed mechanism-based correlations for subcooled flow boiling

CHF. Specifically, Chang (Ref. 37) developed a correlation for CHF based on the idea that large bubbles are broken up in a hydrodynamic limit, thereby reducing the heat-transfer coefficient. Thorgerson, et al., (Ref. 201) proposed a correlation which allowed a burnout condition as a result of a critical friction factor. His correlation falls somewhat in the class of expressions based on boundary-layer separation. Both of these correlations have met with little success in experimental verification or general acceptance.

Probably the most widely verified mechanism-based correlation uses the superposition approach originally proposed by Rohsenow for fully developed nucleate flow boiling and extended to CHF prediction by Gambill (Ref. 202):

$$q''_{CHF} = [q''_{CHF}]_{\text{pool boiling}} + [q''_{CHF}]_{\text{forced convection}} \tag{21}$$

where

$$[q''_{CHF}]_{\text{pool boiling}} = K i_{fg}\, \varrho_v\, [\sigma g_c g\, (\varrho_\ell - \varrho_v)/\varrho_v^2]^{1/4}\, [1 + (\varrho_\ell/\varrho_v)^{0.923}\, (c_p \Delta T_{sub}/25 i_{fg})]$$

in which

$$K = 0.12 - 0.17,\ c_p \text{ is evaluated at } T = T_{sat} - (\Delta T_{sub}/2)$$

$$[q''_{CHF}]_{\text{forced convection}} = h_{conv}\, [(T_w)_{CHF} - T_b]$$

where $(T_w)_{CHF}$ is evaluated with Bernath's (Ref. 203) generalized plot of wall superheat at burnout (Fig. 40). Gambill found that the correlation predicted burnout for a large number of experimental data points within a maximum deviation of 40 percent for various fluids and 17.8 percent for water. The data compared were for numerous fluids and flow configurations with a range of conditions of velocity: 0 to 174 ft/sec, pressure: 4 to 3,000 psia, subcooling (various locations): 0° to 506°F, acceleration: 1 to 57,000 g, CHF: 0.1×10^6 to 37.4×10^6 Btu/hr ft^2. Levy (Ref. 204) presented a similar superposition correlation for CHF:

$$q''_{CHF} = q''_P + q''_C + q''_F \tag{22}$$

where

$$q''_P = 0.131\, i_{fg}\, \varrho_v\, [\sigma g_c^2\, (\varrho_\ell - \varrho_v)/\varrho_v^2]^{1/4}$$

$$q''_C = 0.696\, (k_\ell \varrho_\ell c_\ell)^{1/2}\, [(\varrho_\ell - \varrho_v)/\sigma]^{1/4}\, [\sigma g_c^2 (\varrho_\ell - \varrho_v)/\varrho_v^2]^{1/8}\, \Delta T_{sub}$$

and

$$q''_F = h_\ell (\Delta T_{sat}) + h_\ell \Delta T_{sub}$$

where h_ℓ is determined from a pure forced-convection Nusselt number,

$$h_\ell = Nu\ k_\ell/D$$

and (ΔT_{sat}) is determined by trial and error from a fully developed nucleate boiling relation such as Jens and Lottes [Eq. (11)] at q''_{CHF}.

The list of mechanism-based CHF correlations is by no means limited to those presented here. Those for which the equations are presented appear to be generally applicable to highly subcooled, water flow boiling which will be encountered in the cooling of high-enthalpy arc facility components. A statement made by Bergles in Ref. 8, which is applicable here, is that there are enough "adjustable" constants in any predictive equation to permit an acceptable correlation of data.

4.5.2 Dimensional Analysis/Similitude-Based CHF Correlations

Although dimensional analyses have been performed by Barnett (Ref. 205), Zenkevich (Ref. 206), and Kampfenkel (Ref. 207), Glushchenko (Ref. 86) used partial modeling to obtain a correlation for highly subcooled (greater than 25°C) water flow in tubes and annuli based on the following dimensionless groups:

$$K_1 = q''_{CHF}/i_{fg}\ \varrho_v\ V_{avg}$$

$$K_2 = c_p\ \Delta T_{sub}\ \varrho_\ell/i_{fg}\ \varrho_v$$

$$K_3 = k_\ell/V_{avg} D_e c_p \varrho_\ell$$

$$K_4 = i_{fg}/c_p \Delta T_{sat}$$

Glushchenko proposed the correlation

$$K_1 = 18.25\ K_2^{0.35}\ K_3^{0.5}\ K_4^{1.2} \tag{23}$$

which agreed reasonably well with experimental data (80 percent of nearly 200 selected data points correlated within 25 percent). The data covered the ranges: mass velocity from 500 to 40,000 kg/m^2 sec (100 to 8,200 lbm/ft^2 sec), pressure from 4.9×10^5 to 197×10^5 N/m^2

(72 to 2,900 psia), subcooling (location not discussed) from 25° to 250°C (45° to 450°F), diameter from 2 to 12 mm (0.08 to 0.5 in.), and L/D from 10-20 to 60-120. Ornatskii, et al. (Ref. 208), later extended the approach to low subcooling and moderate quality water flow.

Probably the most successful approach to similitude analysis of the CHF phenomena was performed by Ahmad (Ref. 70). The primary impetus for such analyses was the need for fluid-to-fluid modeling capability. Refrigerants such as Freon have been used in previous experiments (Refs. 66-69) to simulate water cooling at high heat flux conditions, since the CHF for Freon can be achieved at much lower heat flux. Scaling methods were necessary to provide a means of relating the Freon test results to water for high heat flux/water cooling predictions. Ahmad used classical dimensional analysis to identify 12 independent groups, of which 6 were eliminated through inductive arguments. Of the remaining six terms, 3 terms (subcooling number $\Delta i/i_{fg}$, liquid/vapor density ratio ϱ_ℓ/ϱ_v, and L/D) can be independently satisfied in a controlled experiment (controlled inlet temperature, pressure, and geometry). The remaining 3 terms (Reynolds number, Weber-Reynolds number, and liquid/vapor viscosity ratio) are distorted through the liquid and vapor viscosities when the above conditions are fixed. Ahmad solved this problem of multiple distortion by expressing the remaining 3 terms as a CHF modeling parameter ψ_{CHF}, and using a compensated distortion model to relate the terms through two empirical exponents. The final form of the modeling parameter is written as

$$\psi_{CHF} = [(GD/\mu_\ell)\,(\mu_\ell^2 / \sigma D \varrho_\ell)^{2/3}\,(\mu_v/\mu_\ell)^{1/5}] \tag{24}$$

From the Buckingham Pi theorem the dependent parameter, boiling number (q''_{CHF}/Gi_{fg}), is written as follows:

$$q''_{CHF}/Gi_{fg} = f\,(\psi_{CHF},\ \Delta i_i/i_{fg},\ \varrho_\ell/\varrho_v,\ L/D) \tag{25}$$

In addition to the independent parameters identified above, Ahmad included the ratio of the heated equivalent diameter (4 × flow area/heated perimeter) to the hydraulic diameter (4 × flow area/wetted perimeter), D_{he}/D, for more complex geometries. Ahmad demonstrated reasonable scaling of subcooled Freon CHF data from Coffield, et al. (Ref. 209) to water CHF using the above approach. Analysis of complex geometries by this method is limited by the modeling parameter in which the empirical exponents are highly sensitive to geometric parameters. Use of the correlation for situations other than extrapolation of refrigerant CHF data to water CHF has been very limited (see, for example, Katto, Ref. 210).

4.5.3 Empirical-Based CHF Correlations

Probably the most common approach to obtaining a CHF correlation is the curvefitting of experimental test data, although some confusion exists as to whether or not a correlation is considered empirical when some or all of the parameters used in the correlation are obtained from dimensional analysis or limited boiling mechanism analyses. Because most correlations rely on some form of empiricism, this is probably a moot point, and some overlap of the mechanism-based, dimensional analysis/similitude-based, and empirical-based approaches exists. A word of caution about the use of empirical-based correlations: the expressions typically have been derived for a specific range of experimental conditions, and extrapolation of a correlation outside the specific range can lead to very large errors.

The earliest empirical CHF correlations appeared in the late 1940's into the early 1950's, most notably those of McAdams, et al. (Ref. 24), Jens and Lottes (Ref. 88), Gunther (Ref. 27), Buchberg, et al. (Ref. 100), and McGill and Sibbitt (Ref. 211). The correlations proposed by McAdams' team and Gunther are limited to low pressure and only McGill and Sibbitt evaluated subcoolings greater than 250°F. All of the correlations are limited to low coolant velocity (less than 40 ft/sec or a mass velocity less than 2,500 lbm/ft sec). Only the Jens and Lottes correlation included a pressure effect, and none included geometry effects. The test configurations were typically tubes except those used by McAdams' team (annulus) and Gunther (metal strip in a rectangular channel).

Weatherhead (Ref. 43) later modified the Jens and Lottes correlation to include the effects of geometry and latent heat of vaporization. Weatherhead subdivided subcooled flow boiling into four classifications and accounted for variations in boiling characteristics at low and high pressure, low and high mass velocity, small and large geometries, and low and high inlet subcooling. The correlation at high subcooling, however, could only be validated at low L/D, and Tong (Ref. 10) later found that Weatherhead's correlation did not improve the accuracy of prediction when other data were evaluated.

A number of empirical correlations appeared in the late 1950's into the mid- 1960's, many of which are summarized in Ref. 10 (see also Refs. 212 and 213). Most notable are correlations derived by Bernath (Ref. 203), Janssen, et al. (Refs. 214 and 215), Labuntsov (Ref. 216), Van Huff and Rousar (Ref. 29), and the United Kingdom Atomic Energy Establishment at Winfrith, AEEW (Refs. 83, 91, 217, and 218). Bernath developed an empirical correlation based on the premise that at CHF the two-phase flow near the heated surface is highly turbulent and, therefore, well mixed. Assuming the convective heat transfer through this mixture, Bernath empirically formulated expressions for the wall superheat and the heat-transfer coefficient at the critical point (Ref. 219). He later (Ref. 203) refined the correlation for an extended range of geometric variables, and for water is given as

$$q''_{CHF} = h_{CHF} (T_{w,CHF} - T_b) \tag{26}$$

where

$$T_{w,CHF} = 57 \ln p - 54 [p/(p + 15)] - V/4$$

$$h_{CHF} = 19{,}602 [D_e/(D_e + D_i)] + (\text{slope}) V$$

$$\text{slope} = 86.4/D_e^{0.6} \qquad \text{for } D_e < 0.1 \text{ ft}$$

$$\text{slope} = 162 + (18/D_e) \qquad \text{for } D_e \geq 0.1 \text{ ft}$$

where h_{CHF}* is in Btu/ft^{2}hr°C, $T_{w,CHF}$ and T_b are in degrees Centigrade, and D_i is the heated perimeter (inner diameter) in feet divided by pi. Bernath found that the correlation predicted the CHF for nearly 250 forced water flow experiment data points within 16 percent. The range of parameters for the correlation is: pressure from 23 to 3,000 psia, velocity from 4 to 54 ft/sec, subcooling (various locations) from 0° to 615°F, and hydraulic diameter from 0.143 to 0.66 in. Multiple configurations including tubes, annuli, ducts, and ribbons were used in the experiments.

Janssen and others (Refs. 214 and 215) developed rather cumbersome design correlations (presented in Ref. 10) for CHF in subcooled and saturated water flow in tubes and annuli (inner wall heated). Their correlations are valid over the parameter ranges: pressure from 600 to 1,450 psia, mass velocity from 0.2 to 6.2 lbm/ft^2 hr, and hydraulic diameters from 0.25 to 1.25 in. Labuntsov (Ref. 216) assumed (in the first approximation) that the CHF is dependent on pressure, velocity, and subcooling, and developed a CHF correlation for water flow at pressures up to 200 atm. Van Huff and Rousar (Ref. 29) compiled CHF data on 23 different fluids, including water, during the study of heat flux limits of storable propellants. They correlated the higher $V\Delta T_{sub}$ water data as shown in Fig. 7 with the expression

$$q''_{CHF} = 5.1 + 0.000860 \, V\Delta T_{sub} \tag{27}$$

where q''_{CHF} is in Btu/in.^{2}sec. Considerable scatter was noted at low $V\Delta T_{sub}$ (less than 10,000 ft °F/sec), and a significant pressure effect was observed for data obtained in swirl

* Bernath's original work computed h_{CHF} in units of P.c.u./hr ft^2°C, where P.c.u. is a pound-centigrade heat unit. His equations presented here have been converted to Btu rather than P.c.u.

flow. The correlation is valid over the range of parameters: pressure from 10 to 2,000 psia, velocity from 7.5 to 205 ft/sec, and bulk fluid temperature (various locations) from 76° to 470°F.

Lee and Obertelli (Ref. 83) at AEEW in the UK satisfactorily correlated early AEEW subcooled flow boiling CHF data obtained at 1,000 psia using a modified version of an widely used correlation derived by DeBortoli, et al. (Ref. 220) for higher pressure. Lee and Obertelli also found that a more general correlation (based on a local conditions hypothesis) under consideration by AEEW colleagues at about the same period of time (see Barnett, Ref. 221) provided excellent accuracy in the prediction of over 600 CHF data points obtained at AEEW over a range of pressure from 560 to 1,600 psia. Macbeth (Ref. 217) and, later, Thompson and Macbeth (Ref. 218) correlated nearly 4,400 CHF data points (with a root-mean-square error of 7.5 percent) obtained for water flow boiling in round tubes with expressions based on Barnett's local condition hypothesis. The hypothesis assumes that the CHF is only a function of the mass quality at the point of overheating. Macbeth showed that a nearly linear relationship exists at low mass velocity; therefore, he divided the boiling map into a low-velocity regime and a high-velocity regime. Figure 41 presents the boundary limits of the two regimes as identified by Macbeth. The CHF correlations for the two regimes as given by Thompson and Macbeth are

High-Velocity Regime

$$q''_{CHF} = \{[A' + 0.25D\,(G \times 10^{-6})\,\Delta i_i] \,/\, (C' + L)\} \times 10^6 \tag{28}$$

where A′ and C′ are given in Fig. 42,

Low-Velocity Regime

$$q''_{CHF} = \{[(G \times 10^{-6})\,(i_{fg} + \Delta i_i)] \,/[158\,D^{0.1}\,(G \times 10^{-6})^{0.49} + 4L/D]\} \times 10^6 \tag{29}$$

The terms L and D are expressed in inches. Barnett (Ref. 91) later modified the basic form of these equations to obtain a correlation more applicable to annulus data

$$q''_{CHF} = [(A + B\Delta i_i) \,/\, (C + L)] \times 10^6 \tag{30}$$

where

$$A = 67.45\,D_{HE}^{0.68}\,(G \times 10^{-6})^{0.192}\,\{1 - 0.744\exp[-6.512\,D_{HY}\,(G \times 10^{-6})]\}$$

$$B = 0.2587\,D_{HE}^{1.261}\,(G \times 10^{-6})^{0.817}$$

$$C = 185.0\, D_{HY}^{1.415}\, (G \times 10^{-6})^{0.212}$$

$$D_{HE} = (D_O^2 - D_I^2)/D_I$$

$$D_{HY} = D_O - D_I$$

The terms L, D_O, and D_I are expressed in inches. With the correlation above, Barnett correlated 724 annulus CHF data points with a root-mean-square error of 5.9 percent. Limits of the correlation are D_I from 0.375 to 3.798 in., D_O from 0.551 to 4.006 in., L from 24 to 108 in., $G \times 10^{-6}$ from 0.14 to 6.2 lbm/hr ft^2, and Δi_i from 0 to 412 Btu/lbm.

A number of empirical CHF correlations developed for subcooled flow boiling have appeared since the 1960's. Tolubinsky, et al. (Ref. 139), at the Institute of Engineering Thermophysics, USSR, proposed a design equation for determining CHF in annular channels with the inner wall heated, noting that because the CHF is affected by the heated surface shape and dimensions, the CHF data obtained from tube flow cannot be applied to annular flow. Their expression correlated more than 90 percent of approximately 400 experimental data points to within ± 25 percent. The correlation, however, requires knowledge of the CHF during pool boiling conditions for the annular configuration of interest, which may hinder its usefulness. More recent work performed in the USSR regarding CHF in flow boiling correlations includes the studies by Levitan and others (Refs. 222 and 223) and the Scientific Council of the Academy of Sciences of the USSR (Ref. 224). Levitan and his team proposed the following correlation for subcooled and low-quality water flow in tubes:

$$q''_{CHF} = [10.3 - 7.8\,(p/98) + 1.6\,(p/98)^2]\, e^{-1.5x}\, (G/1{,}000)^{1.2\{0.25[(p/98) - 1] - x\}} \quad (31)$$

where q''_{CHF} is in MW/m^2, p is in bars, G is in kg/m^2 sec, and x is the relative quality at CHF. The correlation is good for water flow in circular tubes with a diameter within the range of 4 to 16 mm (0.16 to 0.63 in.). Levitan and his team, along with several other researchers, later assisted the Scientific Council of the Academy of Sciences of the USSR (Ref. 224) in compiling tabular data for predicting CHF in uniformly heated, 8-mm (0.31-in.)-diam tubes. An approximation formula is provided for calculating CHF for tubes whose diameter is different than 8 mm:

$$q''_{CHF} = q''_{CHF,8}\, (8/D)^{0.5} \quad (32)$$

where q''_{CHF} and $q''_{CHF,8}$ are in MW/m^2, $q''_{CHF,8}$ is the value of q''_{CHF} in the 8-mm-diam tube, and D is in millimeters. The tabular data are quoted to have a root-mean-square error of 10 percent. The tabular correlation is limited to the following range of conditions: pressure from 29.5 to 196 bar (435 to 2,880 psia), mass velocity from 750 to 5,000 kg/m^2 sec (154 to 1,024 lbm/ft^2 sec), subcooling (location not discussed) from 0° to 75°C (0° to 135°F), diameter from 4 to 16 mm (0.16 to 0.63 in.), and length-to-diameter ratio of 20 or greater. Doroschuk, et al. (Ref. 222) modified Eq. (31) to include a variable critical pressure such that the correlation could be used for other fluids.

Bowring (Ref. 225) built upon the work of Thompson and Macbeth (Ref. 218) and developed a CHF correlation using four basic variables as functions of pressure. He showed that the correlation predicted approximately 3,800 data points with a root-mean-square error of 6.96 percent. At about the same time, Becker, et al. (Ref. 226) proposed a CHF correlation for round tubes (their data were for primarily 10-mm-diam tubes). Their correlation is given as

$$q''_{CHF} = [G(450 + \Delta i_i)/(40L/D + 156G^{0.45})]\,[1.02 - (p_r - 0.54)^2] \tag{33}$$

where q''_{CHF} is in W/cm^2, D and L are expressed in meters, Δi_i in kJ/kg, G in kg/m^2 sec, and p_r in bars. The expression correlated over 500 data points to a root-mean-square error of 5.7 percent, and is good for the following range of conditions: pressure from 120 to 200 bar (1,765 to 2,940 psia), mass velocity from G(p), as given in Fig. 43, to 7,000 kg/m^2 sec (up to 1,435 lbm/ft^2 sec), inlet subcooling from 8° to 272°C (14° to 490°F), heated length from 2,000 to 5,000 mm (79 to 197 in.), and steam quality from -0.3 to 0.6, where negative quality represents a subcooled liquid.

Knoebel, et al. (Ref. 227) developed a CHF correlation for subcooled water flow based on previous work performed at the Savannah River Laboratory:

$$q''_{CHF} = K\,(1 + 0.0515V)\,(1 + 0.069\Delta T_{sub}) \tag{34}$$

where K equals 153,600 for stainless steel heaters. The correlation is limited to low-pressure water flow (less than 45 psia) and higher subcooling (greater than 45°F). Green and Lawther (Ref. 228) developed a CHF correlation for high-pressure water as well as Freon,

$$q''_{CHF} = Re_V^n\,(\varrho_\ell/\varrho_v)^m\,Pr_V^w\,\sigma_N^p\,f\,(L_{sat}/D)\,(1 + \delta)\,(1 - \delta_\ell) \tag{35}$$

where

$$n = 1 - \exp\,[-0.0067\,(L_{sat}/D)]$$

$$m = 0.1 + \exp\,(-0.007\,(L_{sat}/D)]$$

$$p = -0.5 \{0.15 + \exp [-0.007 (L_{sat}/D)]\}$$

$$w = -\{0.21 + 0.55 \exp [-0.007 (L_{sat}/D)]\}$$

$$f (L_{sat}/D) = 9 \times 10^{-5} \exp \{-0.00055 (L_{sat}/D) + 3.83 \exp [-0.00396 (L_{sat}/D)]\}$$

$$\delta = \exp \{-[0.14 \times 10^{8} \sigma_N + 0.02 (L/D) Pr_v]\}$$

$$\sigma_N = \sigma/(D\Delta i_v \varrho_v)$$

$$\delta_\ell = 0.75 \exp \{-B(L/D) (V_m \sigma/\varrho_\ell \mu_\ell \Delta i_v)\}$$

$$B = 130.5 \exp \{5.0 \exp (-0.02 L/D)\}$$

The expression was quoted to have correlated over 4,250 data points with a root-mean-square error of less than 10 percent, although the correlation proposed by Thompson and Macbeth (Ref. 218) predicted the higher pressure data better. The correlation is limited to reduced pressure less than 0.7, mass fluxes greater than 200 kg/m^2 sec (41 lbm/ft^2 sec), and exit quality greater than 0.1.

Shah presented graphical correlations for CHF in subcooled and saturated flows in tubes (Ref. 229) and annuli (Ref. 230). The tube correlation in functional form is

$$Bo = f_1(x_{in}, L/D) \qquad \text{for } Y < 10^4$$

$$Bo = f_2(x_{cr}, Y, p_r) \qquad \text{for } Y > 10^5 \tag{36}$$

where

$$Y = (GDc_p/k_\ell)\,(G^2/\varrho_\ell^2 gD)^{0.4}\,(\mu_\ell/\mu_v)^{0.6}$$

The graphical expressions correlated 90 percent of 1,271 tube data points within ±30 percent. The data include various fluids including water at reduced pressure from 0.0012 to 0.94, mass flux from 6 to 24,300 kg/m^2 sec (1.2 to 4,980 lbm/ft^2 sec), and inlet quality from -3.0 to positive values. The annulus correlation in functional form is

$$Bo = f_3(x_{in}, L/D_{hp}) \tag{37}$$

where

$$D_{hp} = 4 \times \text{Flow area/Heated perimeter}$$

The graphical expression correlated 88 percent of 825 annulus data points and one noncircular geometry data point within ±30 percent. The data include various fluids including water at reduced pressure from 0.017 to 0.9, mass flux from 100 to 15,780 kg/m^2 sec (20 to 3,230 lbm/ft^2 sec), heated perimeter diameter from 5.3 to 96.3 mm (0.2 to 3.8 in.), gap width from 0.5 to 11.1 mm (0.02 to 0.44 in.), L/D_{hp} from 3.7 to 335, and inlet quality from -3.1 to 0.0.

Finally, Vandervort, et al. (Ref. 80), recently presented an optimized CHF correlation for subcooled forced convection in tubes based on the five parameters of mass flux, subcooling, pressure, diameter, and length-to-diameter ratio. The correlation accounts for cross-coupling of the parameters, and is given as

$$\begin{aligned} q''_{CHF} = {} & 17.05\ [(G')^{0.0732\ +\ 0.2390(D')}] \\ & \times\ [(T')^{0.3060(G')\ +\ 0.001730(T')\ -\ 0.0353\ (D')}] \\ & \times\ [(P')^{-0.1289}] \\ & \times\ [1 + 0.01213(D')^{-2.946\ +\ 0.7821(G')\ +\ 0.009299(T')}] \\ & \times\ [1.540 - 1.280\ (L/D)'] \end{aligned}$$

where

$$G' = 0.005 + G/10^5$$

$$T' = 5 + \Delta T_{sub}$$

$$P' = (0.0333 + p)/3.0$$

$$D' = D/0.003$$

$$(L/D)' = (L/D)/40.0$$

and q''_{CHF} is in W/m^2, G is in kg/m^2-sec, ΔT_{sub} in degrees Centigrade, p is in MPa, and D is in meters. The expression correlated 85 percent of over 800 data points within ±25 percent.

$$q''_{CHF,max} = (GDi_{fg}/4L)\ [1 + c_\ell\ (\Delta T_{sub})_i/i_{fg}] \tag{41}$$

Gambill and Lienhard (Ref. 231) presented an interesting semiempirical approach to determining a practical upper limit to CHF. From kinetic theory they derived an expression for $q_{max,max}$, the highest heat flux that can conceivably be achieved in a phase-transition process:

$$q''_{max,max} = \varrho_g i_{fg}\ (RT/2\pi)^{1/2} \tag{42}$$

A practical upper limit to heat transfer by phase change was shown to be $0.1q_{max,max}$; however, the method is limited to pressures less than about one-tenth of the critical pressure. As a point of interest, the highest CHF achieved for any configuration and test condition was nearly 3.4×10^8 W/m^2 (30,000 Btu/ft^2 sec) measured by Ornatskii and Vinyarskii (Ref. 232) with a nonuniformly-heated (circumferential), small-bore tube.

4.6 COMPARISON OF FLOW BOILING AND CHF CORRELATIONS

Guglielmini, et al. (Ref. 172) presented an interesting comparison of flow boiling correlations including boiling incipience and partial nucleate boiling. Most of the boiling incipience correlations, when compared to representative data, had an accuracy within 30 percent. A quantitative comparison of the partial nucleate boiling expressions could not be performed because of lack of data. The correlations for fully developed nucleate boiling generally predicted representative data to within ± 30 percent, although in some cases the data scatter caused error in the prediction to be as large as 70 percent.

Gambill (Ref. 233) compared ten CHF correlations for water flow in a tube (diameter of 0.1 in. and length of 20 in.) at a pressure of 600 psia and subcooling of 100°F over a range of velocity from 10 to 80 ft/sec. Prediction of CHF at the higher velocity using the correlations varied by nearly a factor of four and by a factor of two at the lower velocity. Factors of two or greater between correlation predictions at higher mass velocity have also been noted by Zeigarnik, et al. (Ref. 96) and Boyd (Ref. 17), and the disagreement tends to worsen as mass velocity increases. It should be reiterated that empirical CHF correlations typically have been derived for a specific range of experimental conditions, and extrapolation of a correlation outside the specific range can lead to very large errors. In addition, a given analytical CHF correlation typically has enough "adjustable" constants in the predictive equations to permit an acceptable correlation of a specific data set; however, general application of the correlation to various configurations or test conditions may result in significant errors.

5.0 APPLICABLE EXPERIMENTAL APPROACHES AND MEASUREMENT TECHNIQUES FOR BOILING HEAT TRANSFER

A representative list of flow boiling experiments that have been performed within the last 50 years is presented in Appendix B, and a few comments are in order. The heating method of choice is electrical resistance heating (Joule heating). A fewer number of experiments made use of cartridge heaters, and very few incorporated some other means of heating. The test sections were primarily tubes, although a significant number of experiments had annular test sections. Of the more than 175 listed experiments where water was used, 46 had a maximum heat flux greater than 1 x 10^7 W/m^2 (1,000 Btu/ft^2 sec), and only 10 had a heat flux greater than 5.5 x 10^7 W/m^2 (5,000 Btu/ft^2 sec), with most of these performed at a pressure less than 35 bar (500 psia), subcooling less than 170°C (300°F), or mass velocity less than 30 Mg/m^2 sec (6,200 lbm/ft^2 sec). Clearly, data obtained at high heat flux, pressure, subcooling, and velocity are severely limited.

A number of measurement techniques used in these and other experiments are worth reviewing. One of the most important measurements in a boiling experiment is surface temperature. Placement of a discrete, intrusive temperature measuring device such as a thermocouple on the surface at the fluid interface may disrupt the flow field, causing large errors in the temperature measurement. For this reason, backside wall temperature is typically measured and used to estimate surface/fluid interface temperature through the use of an analytical temperature distribution. On an electrically heated surface, the temperature measurement with a thermocouple (which provides temperature from millivolt changes) is further complicated by voltage drop in the test specimen due to current flow. Hughes (Ref. 21) incorporated a single wire junction at the surface and the two-wire thermocouple junction formed 0.8 mm (1/32 in.) from the surface, thereby preventing any extraneous voltages due to voltage drop between the thermocouple leads. Numerous researchers have used thin sheets of mica or other electrical insulating material to isolate the thermocouple from the current-carrying surface. One of the most attractive approaches as suggested by Dutton and Lee (Ref. 234) involves the use of a three-wire thermocouple. The thermocouple circuit is "balanced" by applying and subsequently reversing the current and nullifying the voltage drop with a voltage-dividing potentiometer, the net effect being the same as welding all three wires to the location of the middle wire.

Burnout detectors have been used by a number of investigators (e.g., Refs. 21, 235, and 236) to protect the test sections from destruction as the CHF is approached. However, as pointed out previously, Hughes (Ref. 21) noted that the speed and intensity of the transition to film boiling (i.e., burnout) was observed to increase with increasing velocity and subcooling, and in some cases burnout has occurred instantaneously (Ref. 147). Therefore, the effectiveness of a burnout detector to protect the test section in highly subcooled, high-velocity, high-pressure flow is questionable.

Gunther (Ref. 27) was one of the earliest to perform a detailed photographic study of subcooled flow boiling at higher heat flux, although pressure was limited to below 11.3 bar (164 psia) and velocity limited to below 12 m/sec (40 ft/sec). His photographic system was capable of 20,000 frames/sec and gave a resolution of bubbles as small as 0.1 mm (0.004 in.) diameter. Fiori and Bergles (Ref. 20) used several photographic techniques, including high-speed photographs, movies, and video. They successfully identified cyclic variations in surface temperature from variations in the flow structure by simultaneously recording surface temperature and high-speed photographs of a specific surface location. Ram, et al. (Ref. 237), although studying bubble generation in pool boiling of water, analyzed intensity fluctuations registered by a photomultiplier tube to determine bubble size and growth cycle. Lineberger (Ref. 238) used a dual-frequency sound field to determine bubble size in pool boiling of water. Brown (Ref. 40) used reflected light from bubbles detected by a light-dependent resistor to identify the presence of first vapor when boiling begins.

Several approaches have been used to determine the vapor thickness or void fraction near the heated surface in local boiling. Costello (Ref. 239) measured the number of beta particles emitted from a vial containing radioactive strontium 90 to determine the vapor thickness in a low-velocity, water flow boiling apparatus with an annular test section. Rogers, et al. (Ref. 240) used a traversing gamma-ray densitometer (cobalt-57 source) to measure void fractions in water flow boiling in an annulus. Similar densitometers have been used by Buchberg, et al. (Ref. 100) and Edelman, et al. (Ref. 241) to measure the void fraction of subcooled water flow in tubes. Jiji and Clark (Ref. 242) used a specially developed traversing thermocouple probe to measure bubble boundary-layer thickness and temperature profiles in low-velocity, subcooled water flow. Later, Stefanovic, et al. (Ref. 243) used a similar probe to measure temperature fluctuations within a superheated boundary layer adjacent to a heated annular wall in forced water flow.

The measurement of acoustic noise due to boiling has permitted various researchers to study hydrodynamic instabilities in heated channels and to diagnose the different regimes of boiling. Pressure transducers have typically been incorporated in an experimental apparatus where flow oscillations are to be studied (e.g., Refs. 244 and 245), although Romberg and Harris (Ref. 246) used piezoelectric accelerometers to measure vibrations associated with parallel channel, density wave, and acoustic oscillations in a flow boiling loop. Several investigations have been performed to show that information on the boiling mode can be ascertained from boiling noise (Refs. 247-251). Hydrophones (i.e., transducers) with piezoceramic sensing elements have typically been used to measure the acoustic emissions from boiling.

A number of important considerations can be learned from the success or failure of previous experiments and associated analyses. Westwater (Ref. 11) pointed out that one basic method of proving the correctness of instrumentation in these types of experiments which is widely overlooked is a simple energy balance of the system. Accuracy of the all-important wall temperature can be severely affected by overlooked items such as conduction effects or temperature-dependent material properties. Flow loop layout can affect the hydrodynamic stability of the system, which can seriously degrade the heat-transfer and CHF measurements. Gas content and cooling fluid impurities have been shown to have an effect on boiling heat transfer and CHF, yet in many experiments these have been ignored or have not been adequately quantified. As shown in Appendix B, many researchers fail to present basic test condition values such that their data may be used in other analyses. In many cases, the location in which the subcooling and pressure are measured or determined is not given. Finally, the geometric scale of a heater surface has been shown to be very important. Specifically, Bakhru and Lienhard (Ref. 252) demonstrated that the hydrodynamic processes that give rise to burnout cease to occur when the Laplace number*, R′, is reduced below a value on the order of 0.1 (see also Refs. 33 and 253). Generalized correlations based on data acquired from a configuration with a Laplace number below this value may have very large errors when applied to scaled-up or different configurations.

6.0 SUMMARY

An extensive review of backside water cooling processes that are applicable to high-enthalpy facility components has been conducted. The processes can be identified on the basic boiling curve which, for the configurations of interest, includes the pure convection regime, the nucleate boiling regime, and the boiling crisis point where the CHF occurs. More than 20 parameters have been shown to affect various portions of the boiling curve, and optimum cooling (and greater CHF) occurs at high subcooling and high mass velocity. The primary parameters which affect steady-state boiling heat transfer are pressure, mass velocity, subcooling, heater diameter, and heater length (Ref. 7); however, other parameters have been experimentally shown to have significant effects. Several heat-transfer enhancement techniques have also been reported.

Because of the complexity, the unsteady nature, and the small scale of the flow boiling processes, no general theory has been developed. Reasonable confidence in correlating heat-transfer data in the pure convection and nucleate boiling regimes is shown. The more recent pure forced-convection correlations of Petukhov [Eq. (5)] for turbulent flow in a tube and Kays and Leung (Ref. 153) for turbulent flow in an annulus are probably the most useful for the nonboiling regime. Bergles and Rohsenow's partial nucleate boiling expression [Eq. (8)] best correlates the transition from pure convection to fully developed nucleate boiling,

* The Laplace number is defined as $R' = r[g(\varrho_\ell - \varrho_v)/\sigma]^{1/2}$, where r is a characteristic radius dimension of the heater (see Ref. 252).

and the widely accepted Rohsenow relation [Eqs. (16) and (17)] ranks among the best correlations for fully developed nucleate boiling in forced flow. Considerable disagreement between the large number of analytical and empirical CHF correlations currently exists. Moreover, because the CHF correlations are sensitive to the range of conditions for the data used, no single correlation can be recommended. Those CHF correlations developed with data from high mass velocity, pressure, subcooling, and heat flux experiments using water as the coolant are more appropriate for use in the design of high-enthalpy arc heater components. Recommended correlations in this group include Bernath (Ref. 203), Van Huff and Rousar (Ref. 29), Yagov and Puzin (Ref. 195), Levy (Ref. 204), Labuntsov (Ref. 216), and Vandervort, et al. (Ref. 80). Additional correlation development will be required to account for factors such as acceleration or transient effects at these extreme conditions.

Very few data exist that were obtained at high heat flux, high subcooling, high mass velocity, and high pressure, which are optimum for cooling high-enthalpy facility components. Finally, numerous shortcomings of previous experiments have been identified, and care must be taken in any experimental program where boiling heat-transfer data are acquired for future analysis and correlation.

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from Ref. 3

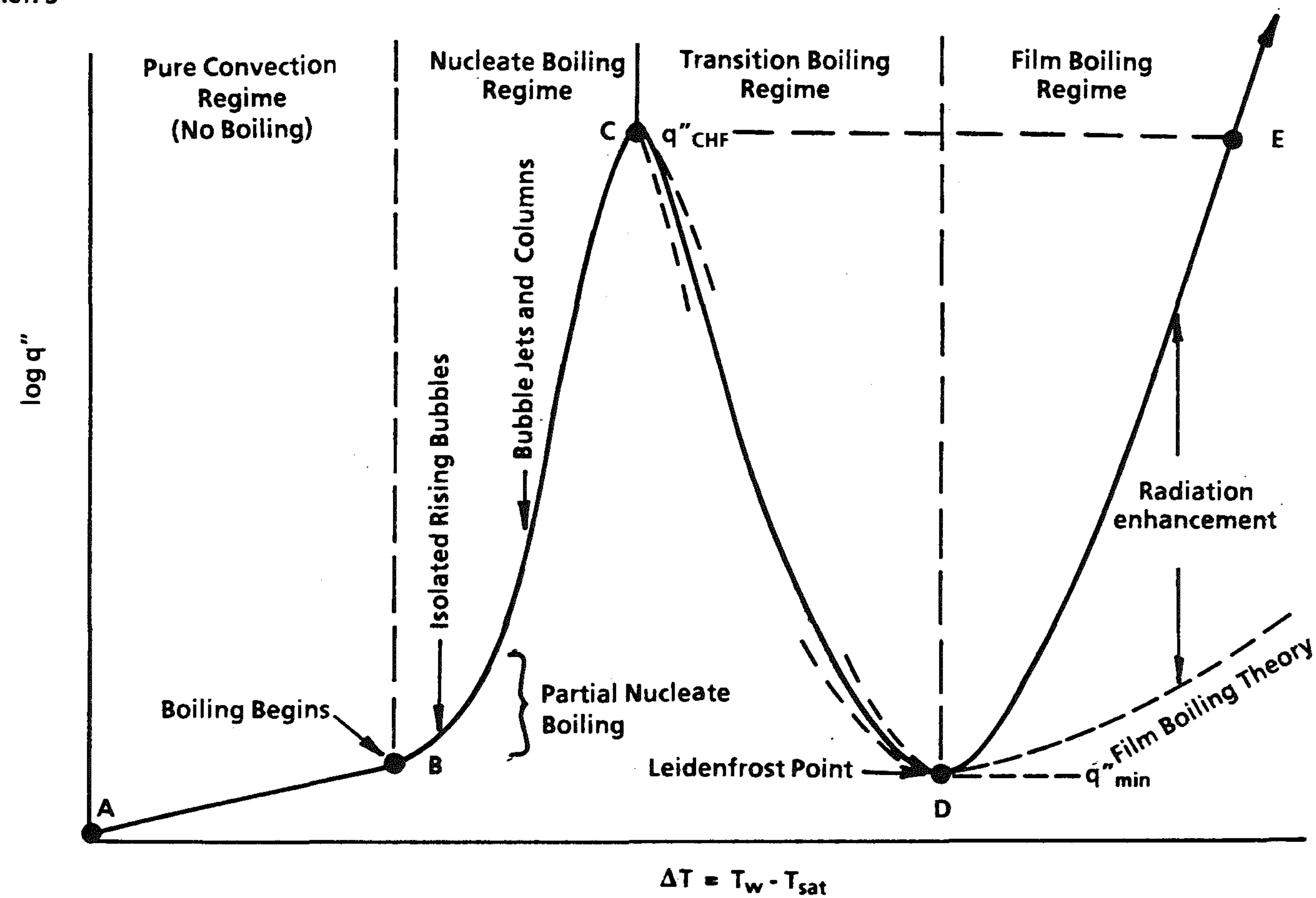


Figure 1. Basic boiling curve.

from Ref. 22

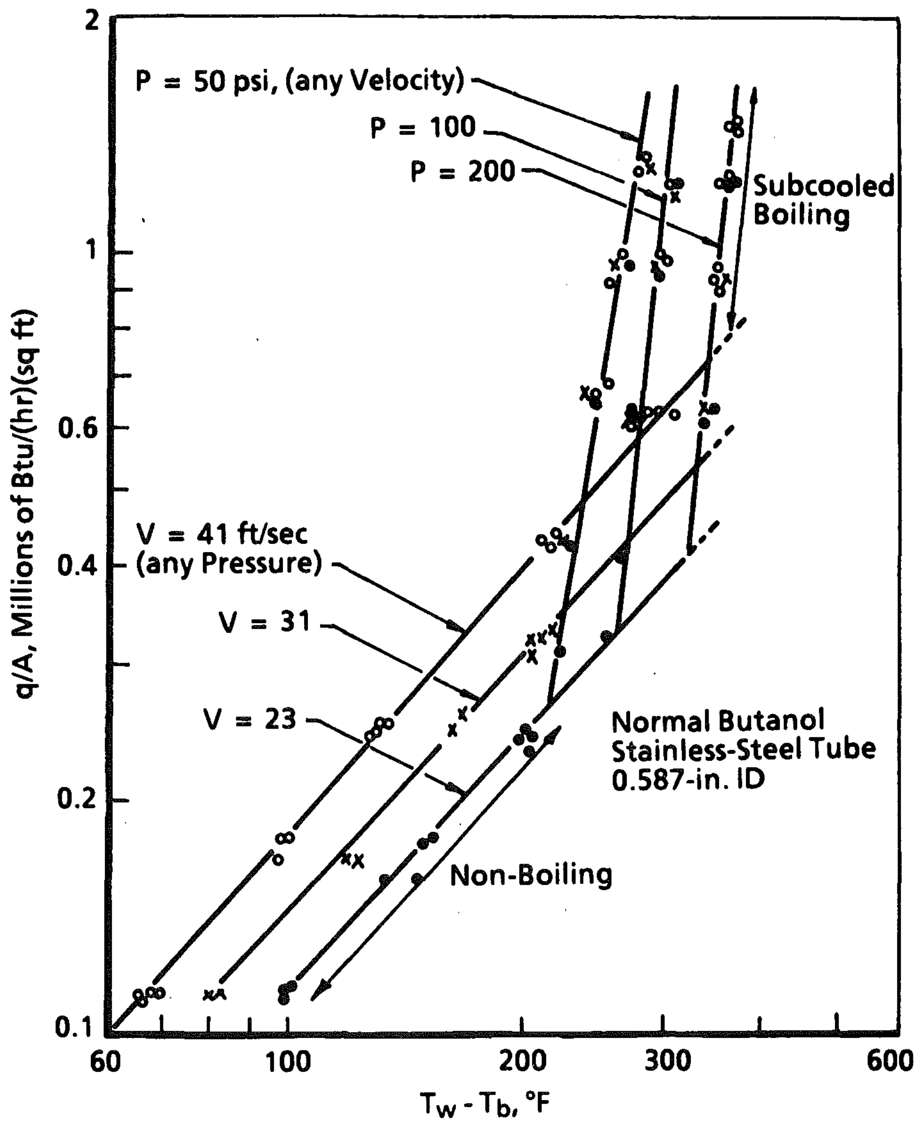
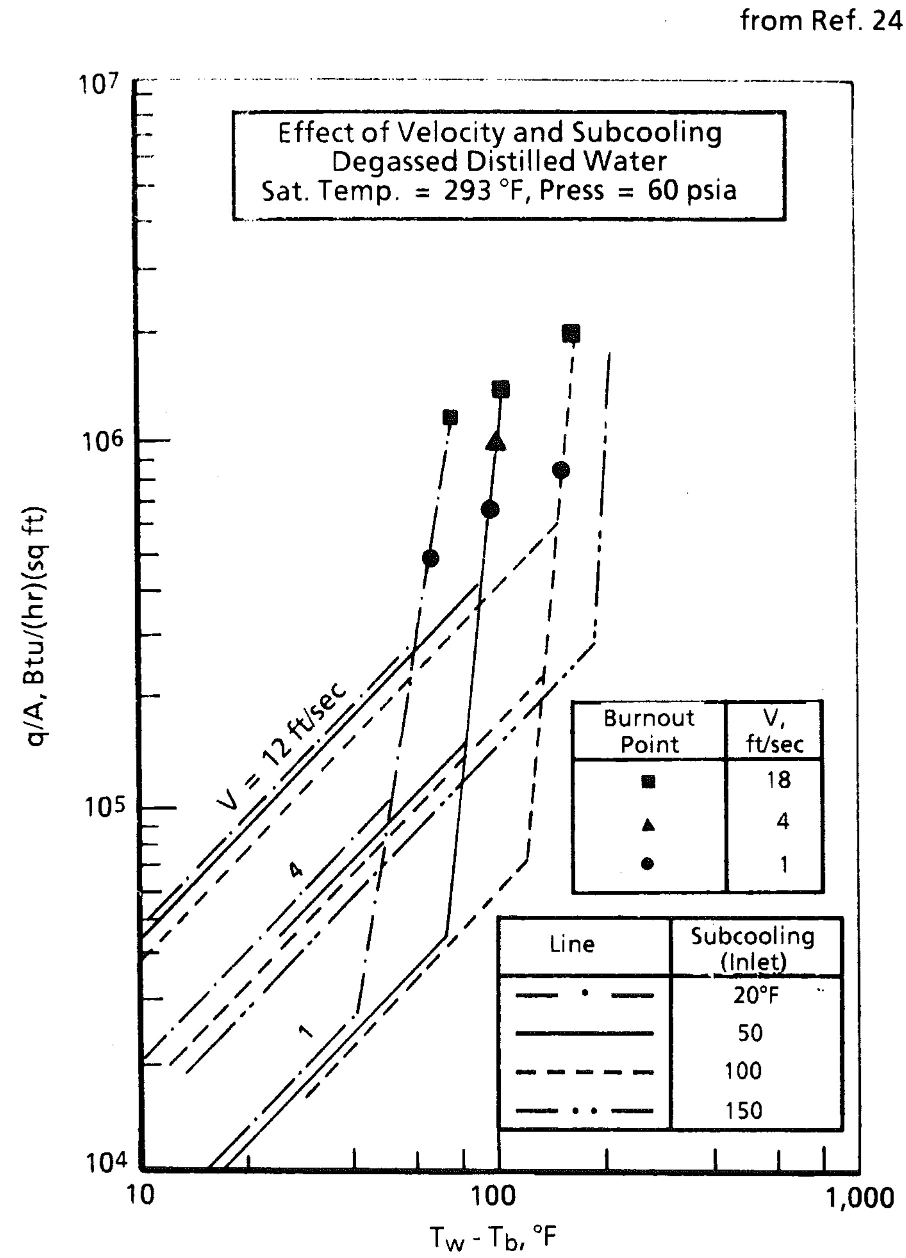


Figure 2. Effect of pressure and velocity on subcooled boiling.



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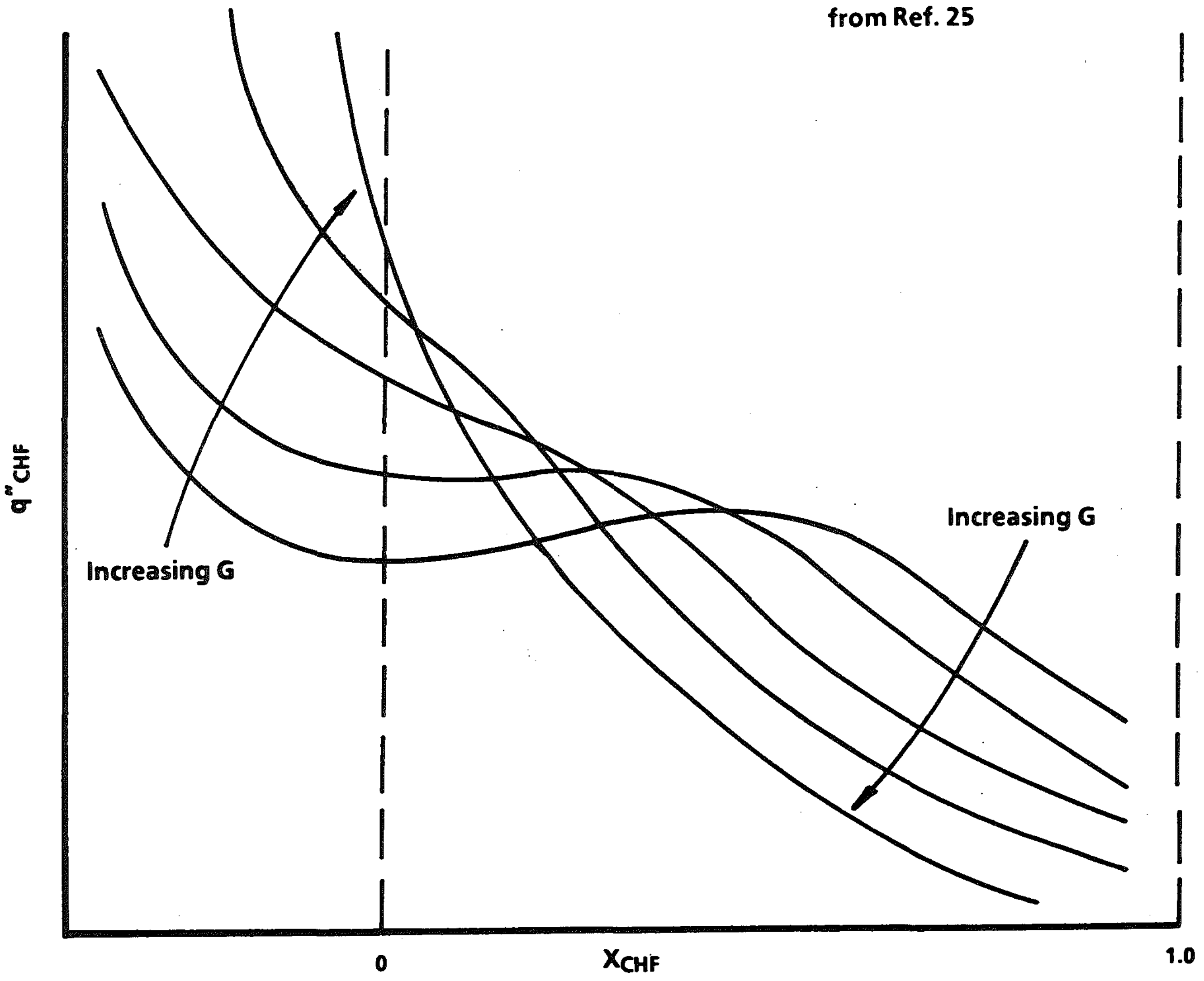


Figure 4. Effect of mass velocity and vapor quality.

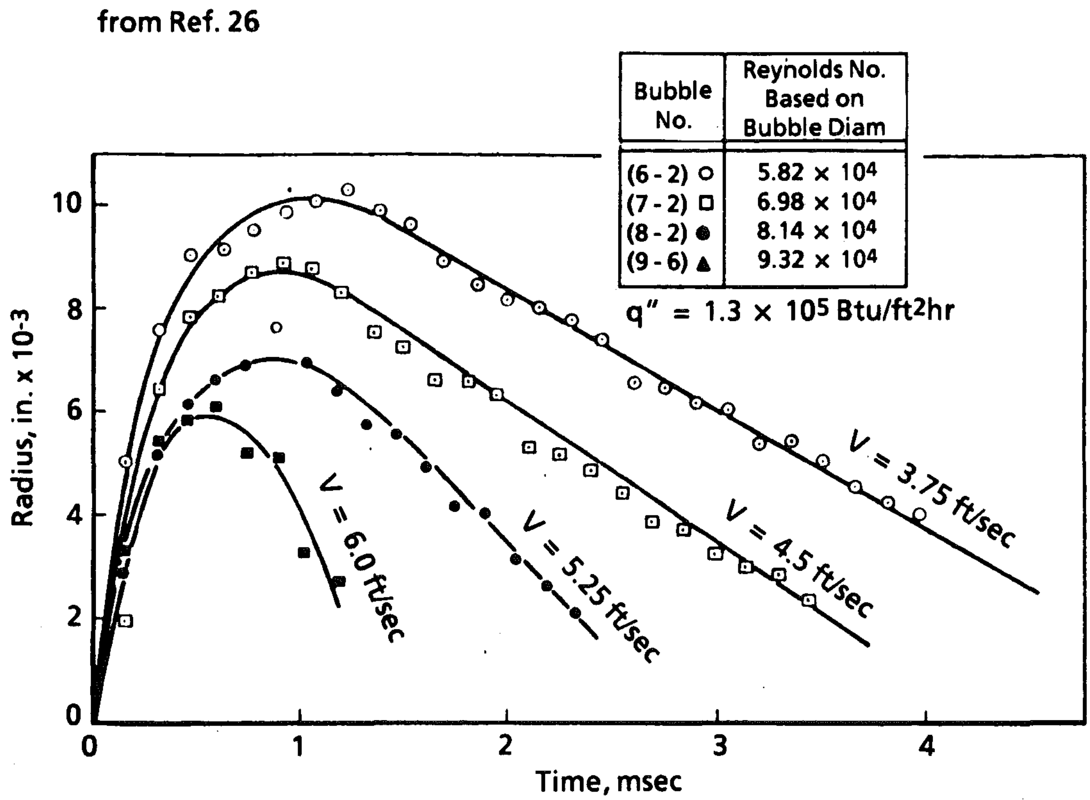


Figure 5. Effect of liquid velocity on bubble growth and collapse.

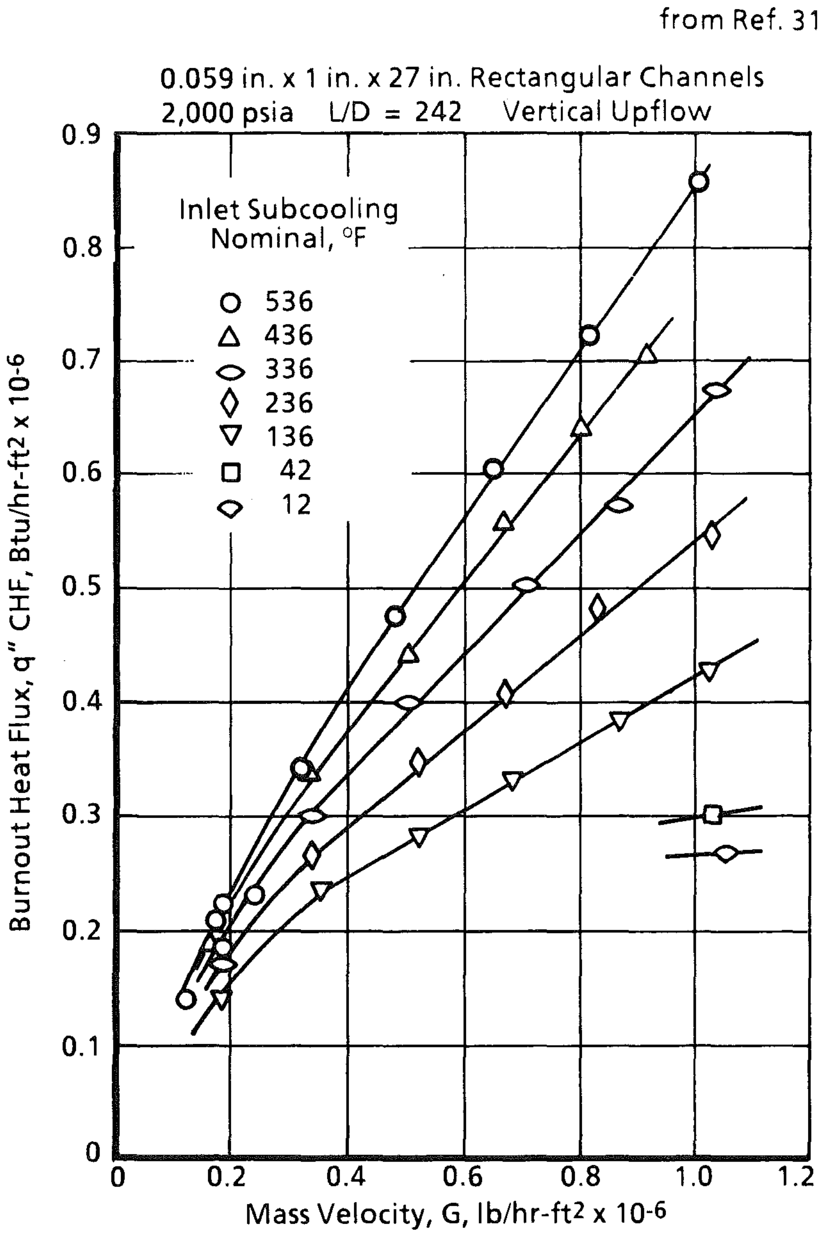


Figure 6. Effect of subcooling with increasing mass velocity.

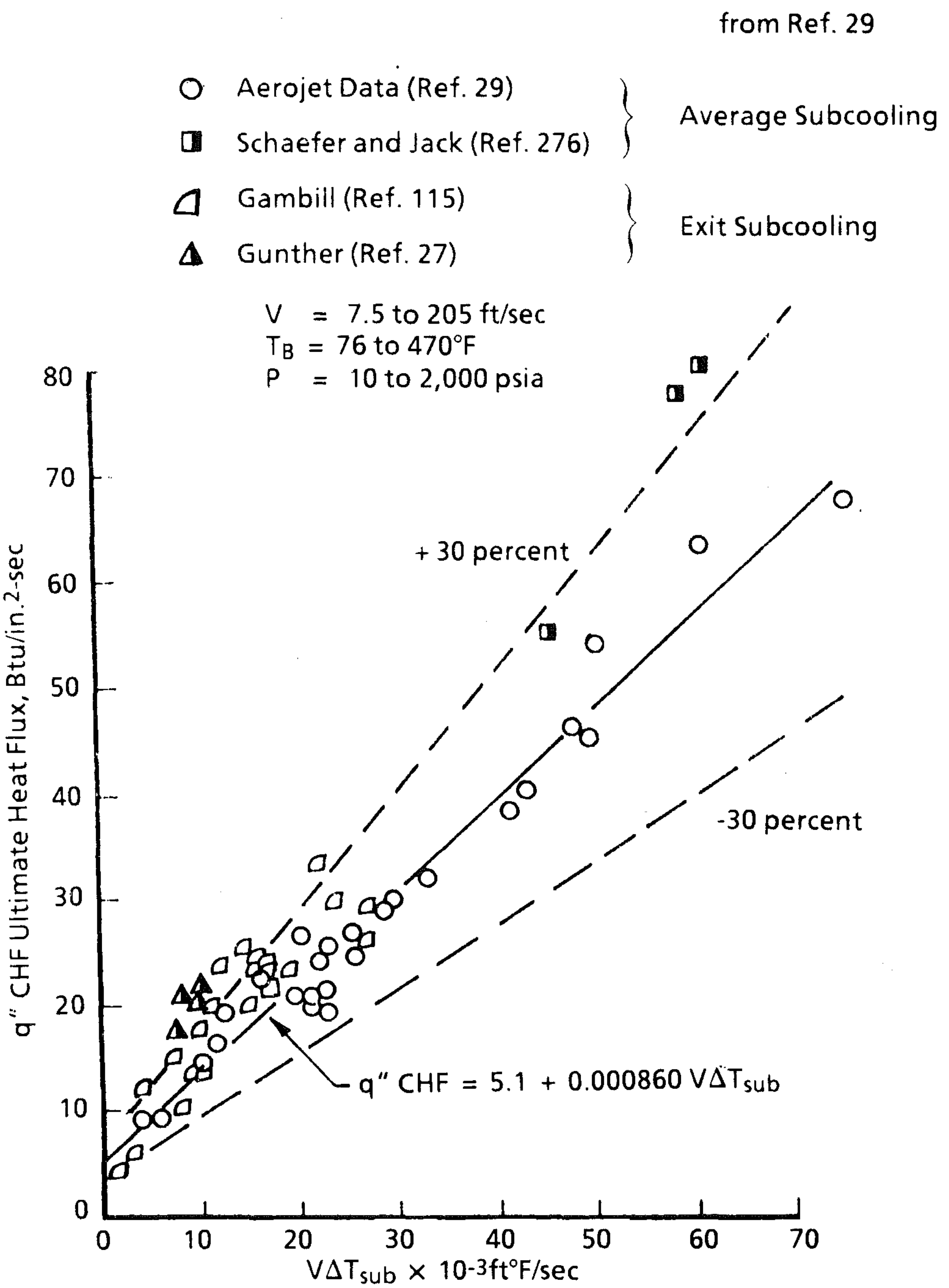


Figure 7. Subcritical pressure water burnout heat flux.

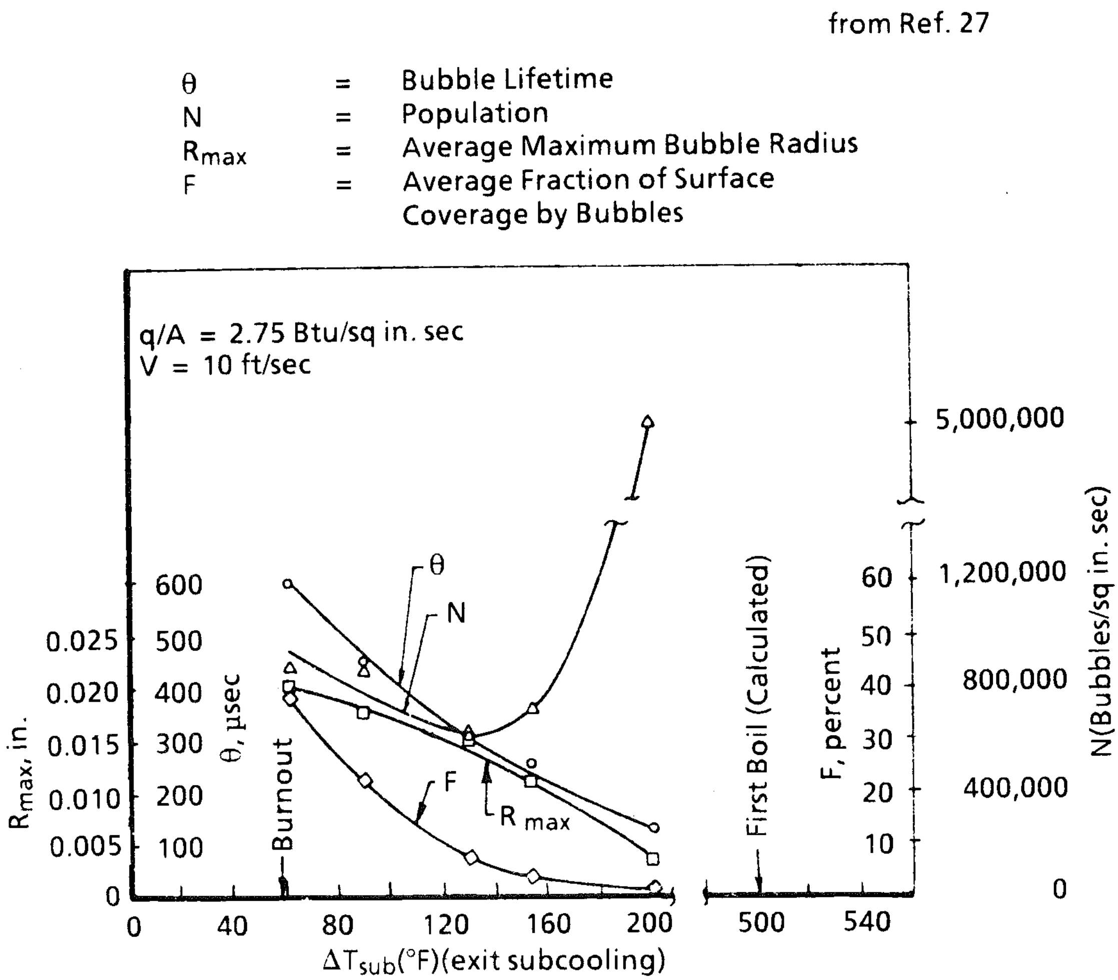


Figure 8. Effect of subcooling on bubble characteristics.

from Ref. 36

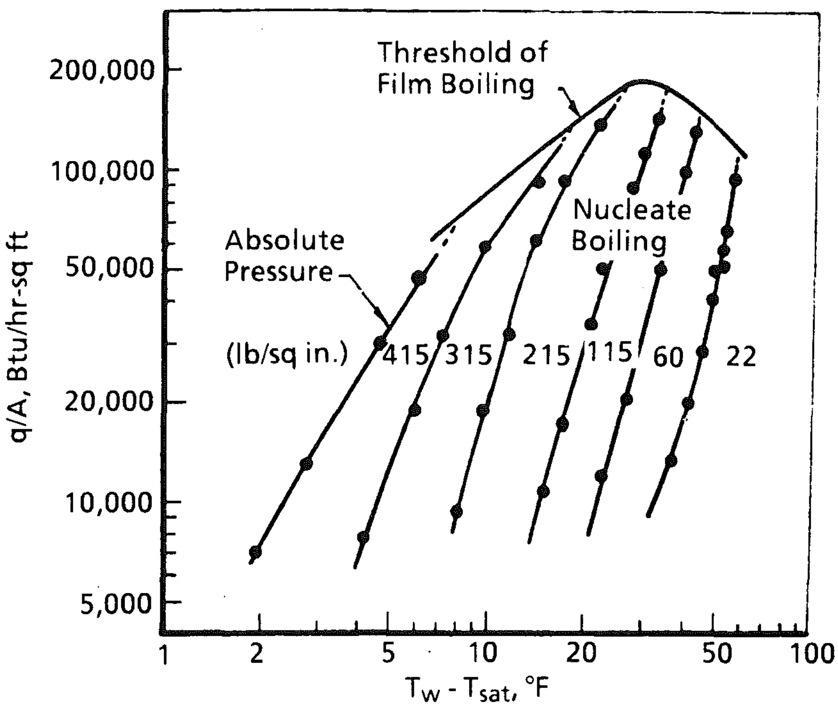


Figure 9. Optimum pressure for the boiling of n-pentane.

from Ref. 15

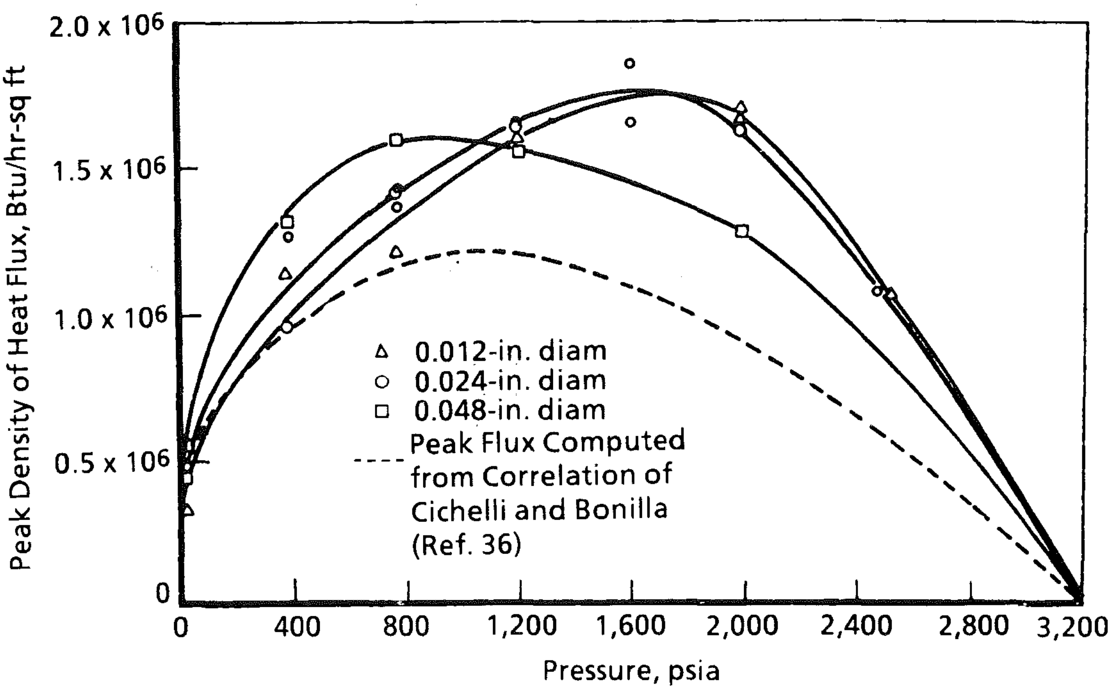


Figure 10. Effect of heater size on optimum pressure.

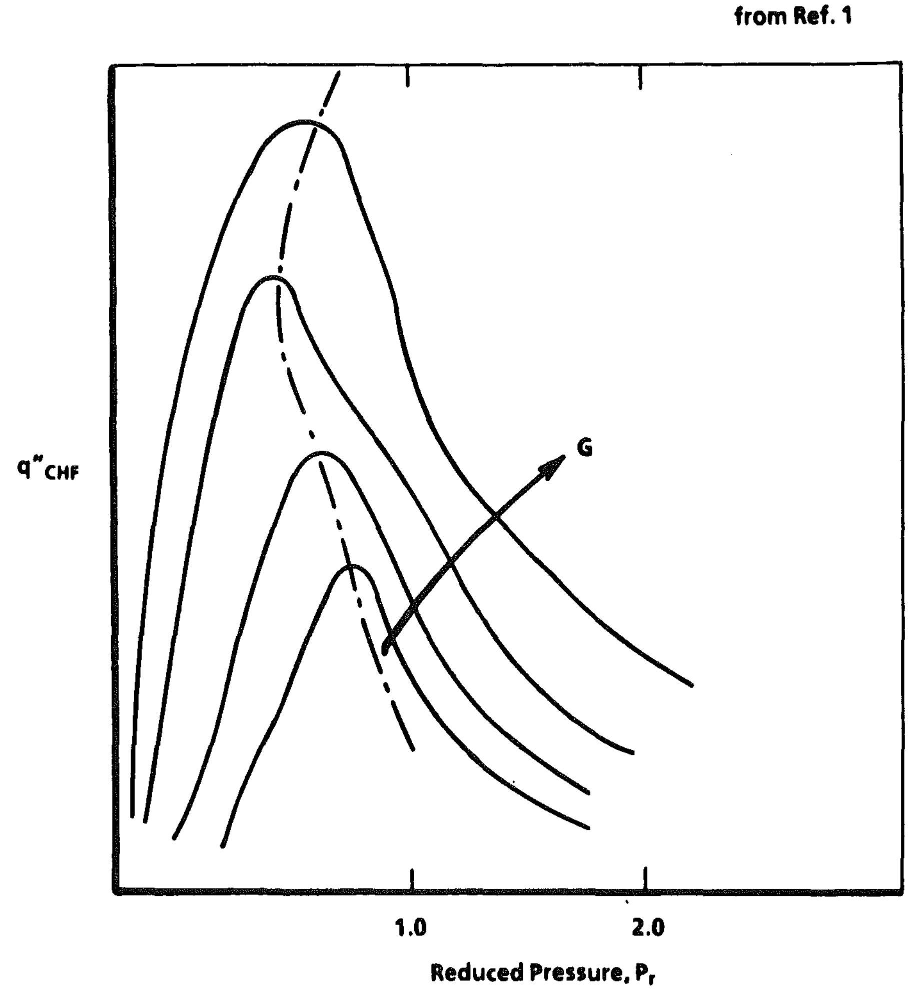


Figure 11. Effect of mass velocity on optimum pressure.

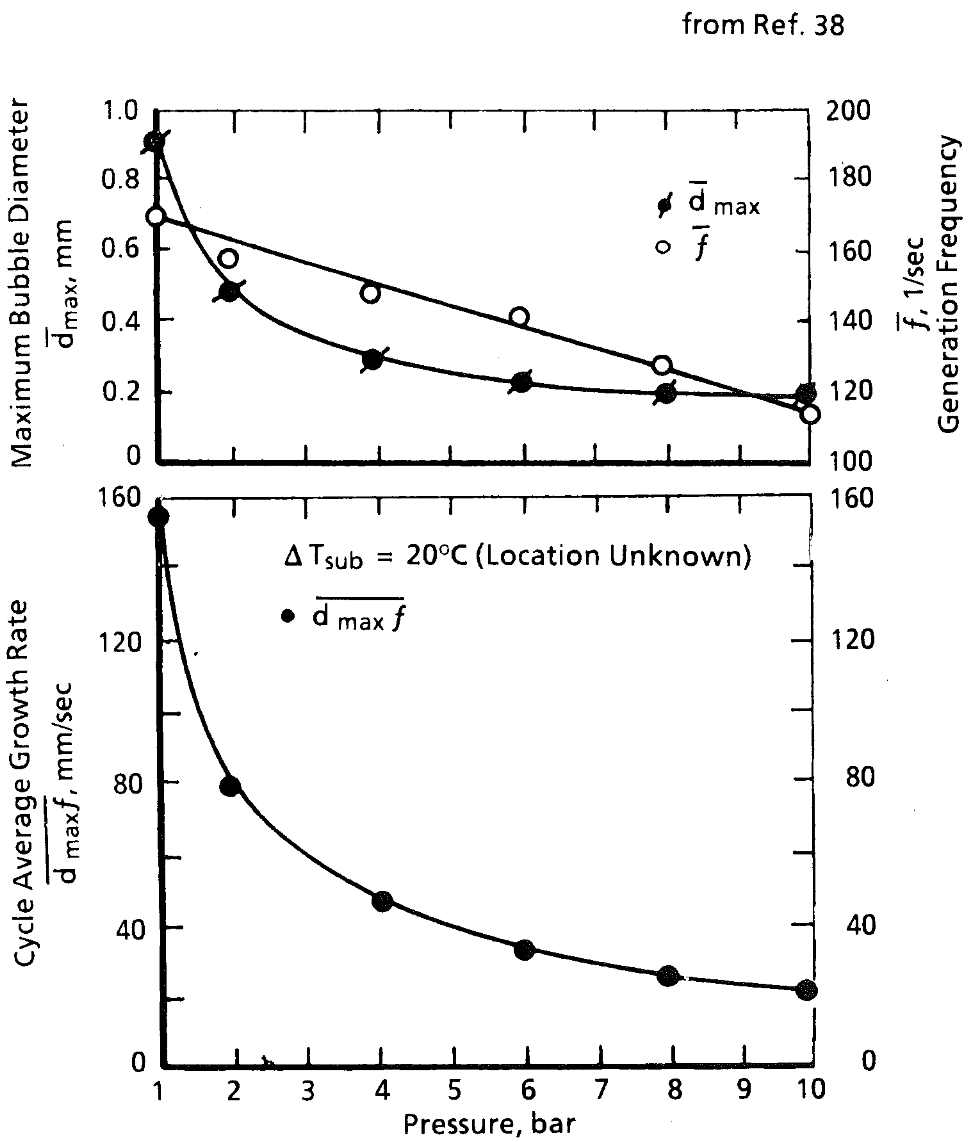


Figure 12. Effect of pressure on bubble characteristics.

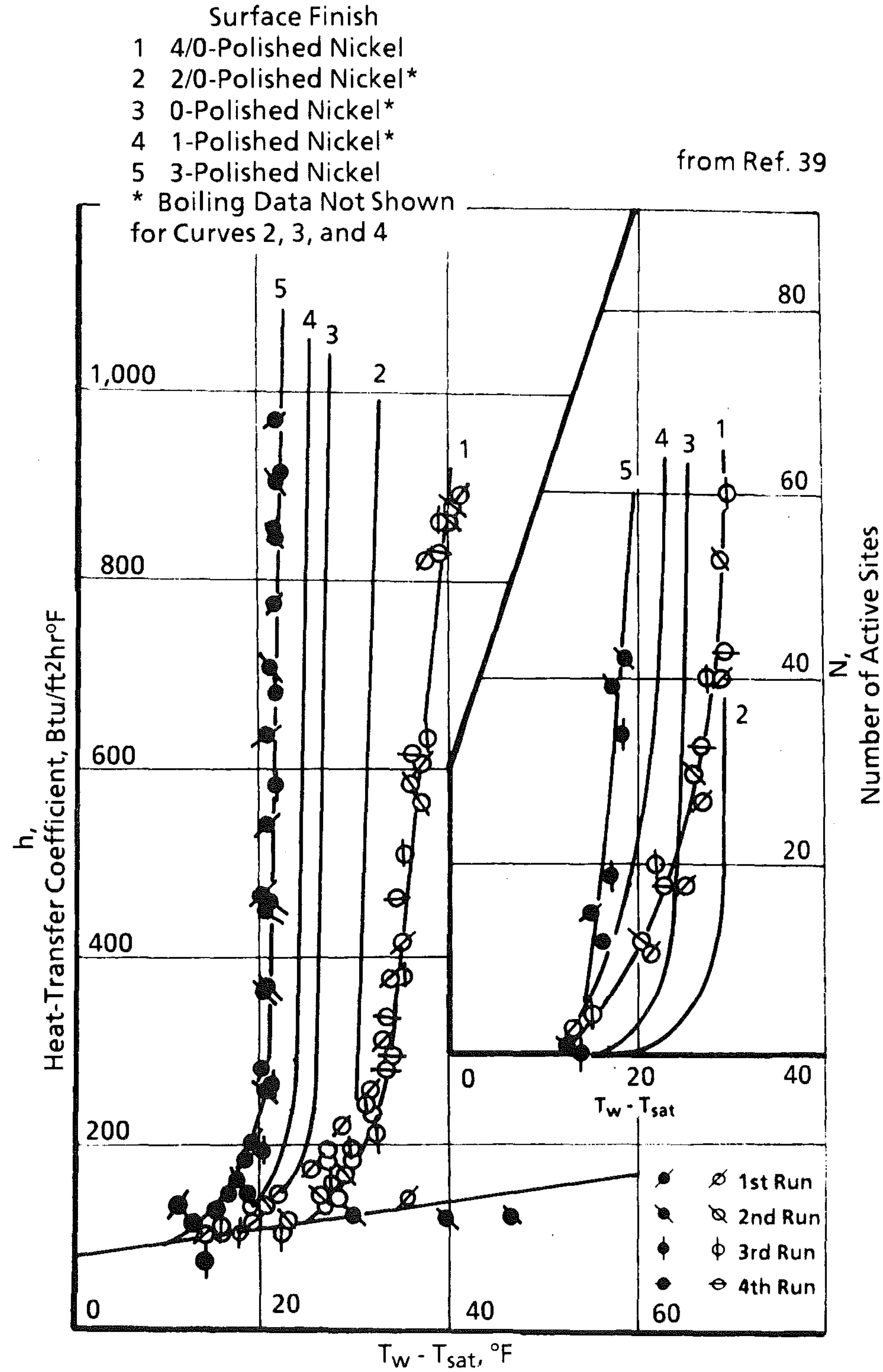


Figure 13. Effect of surface roughness on heat transfer.

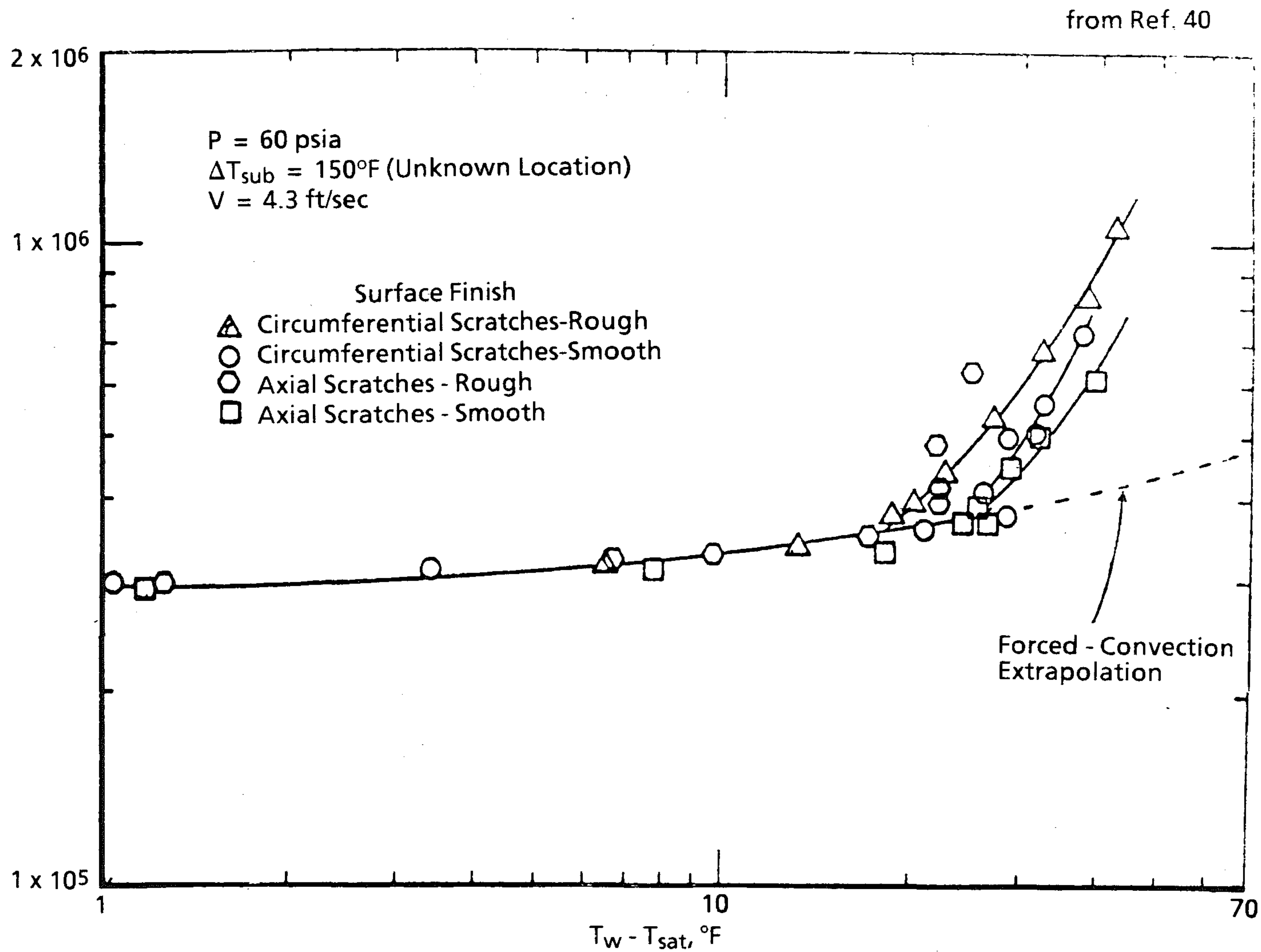


Figure 14. Roughness effect on heat transfer.

from Ref. 47

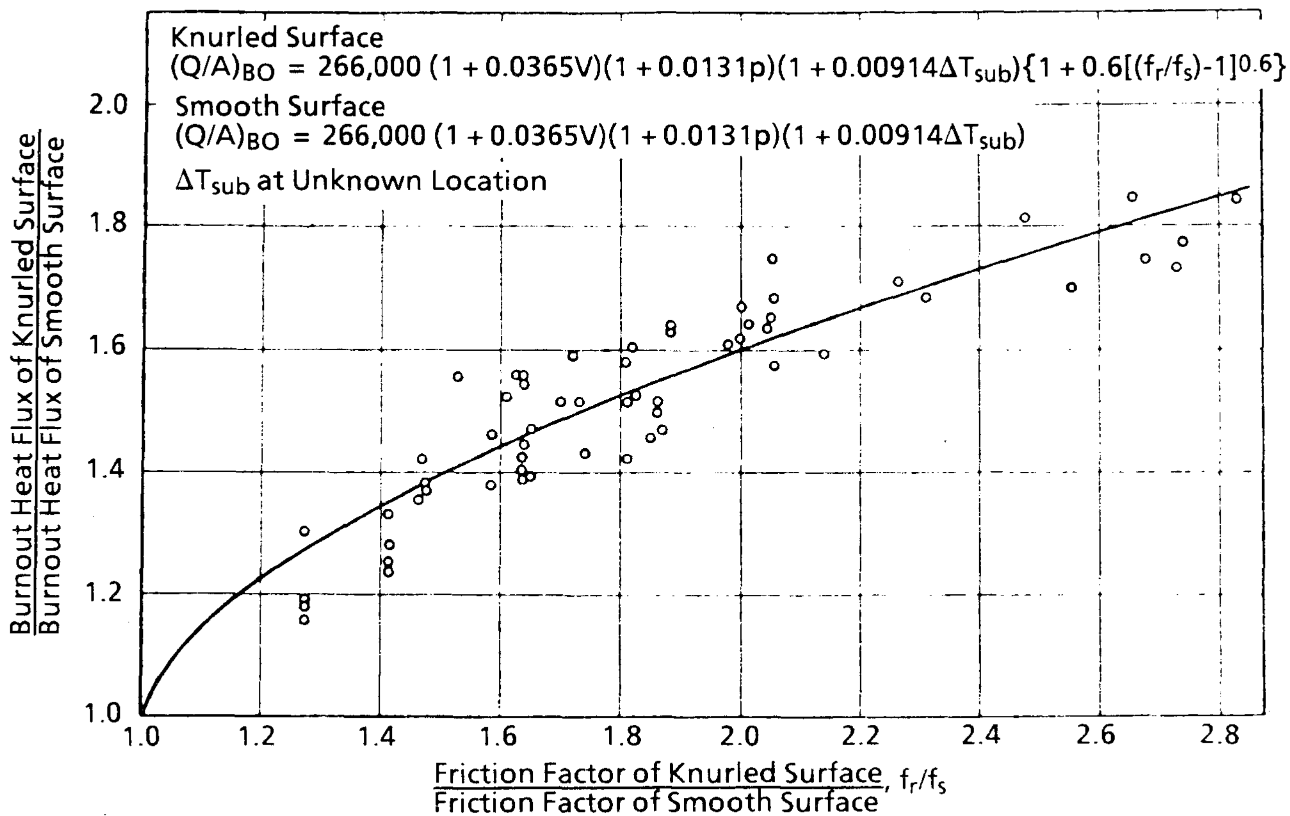


Figure 15. Burnout heat flux of knurled surfaces.

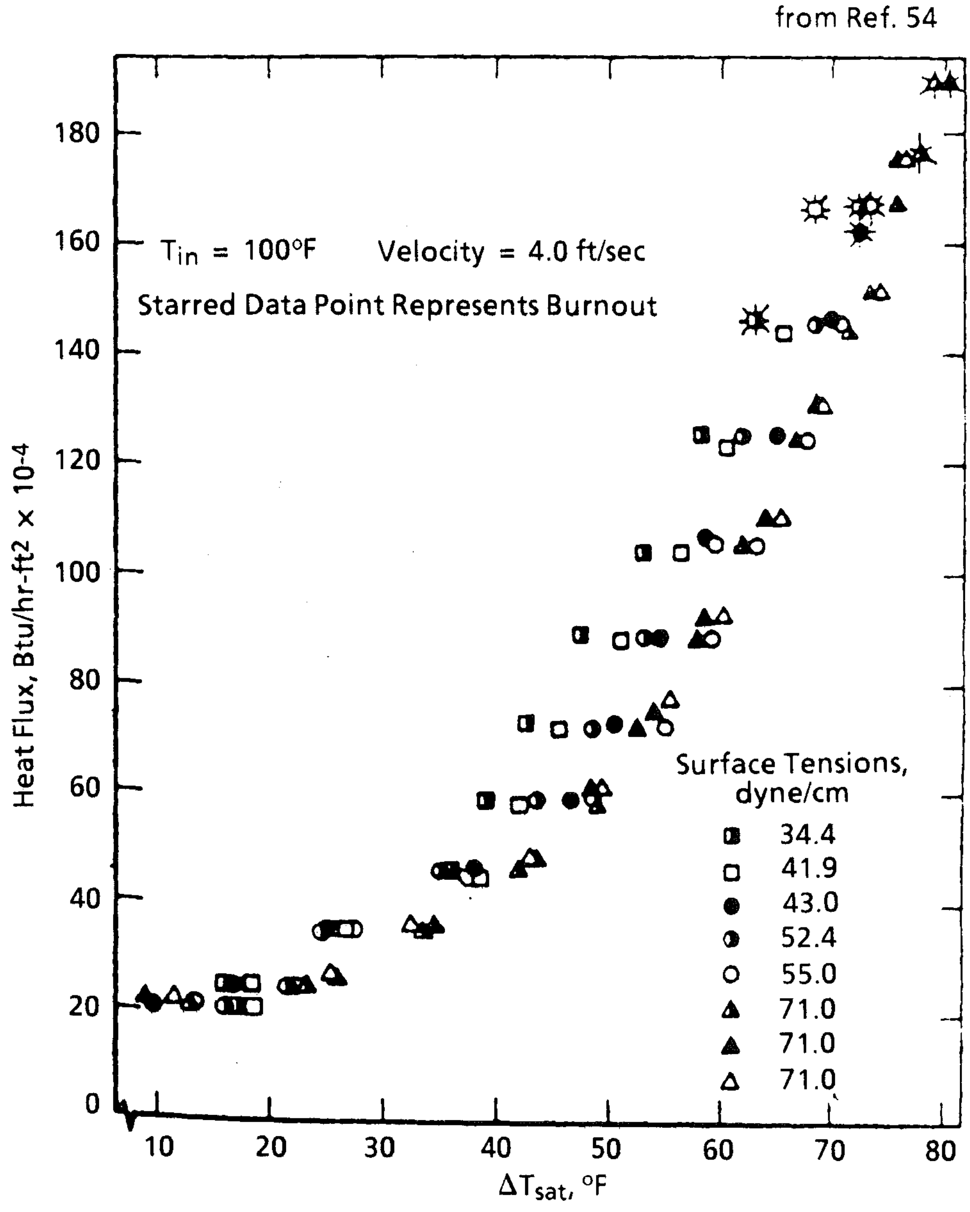
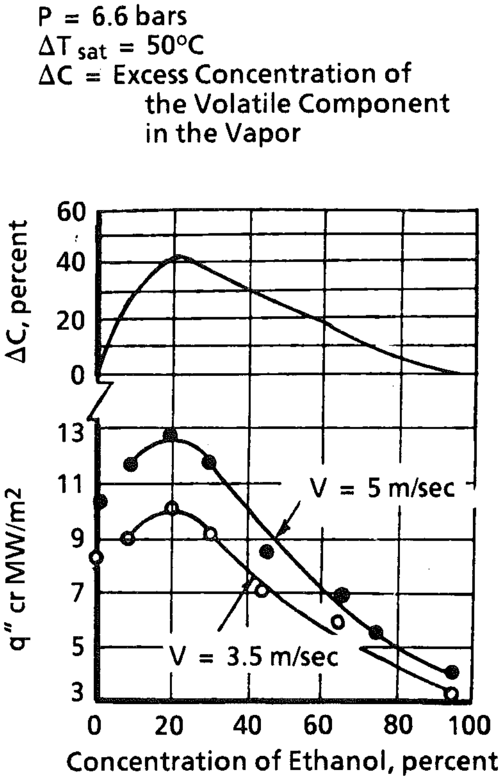
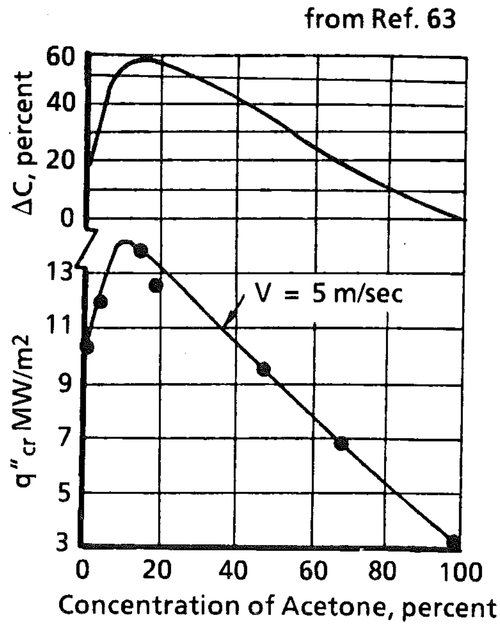


Figure 16. Effect of a surface active agent on heat transfer.



a. Ethanol-water



b. Acetone-water

Figure 17. Effect of binary mixture concentrations on critical heat flux (CHF).

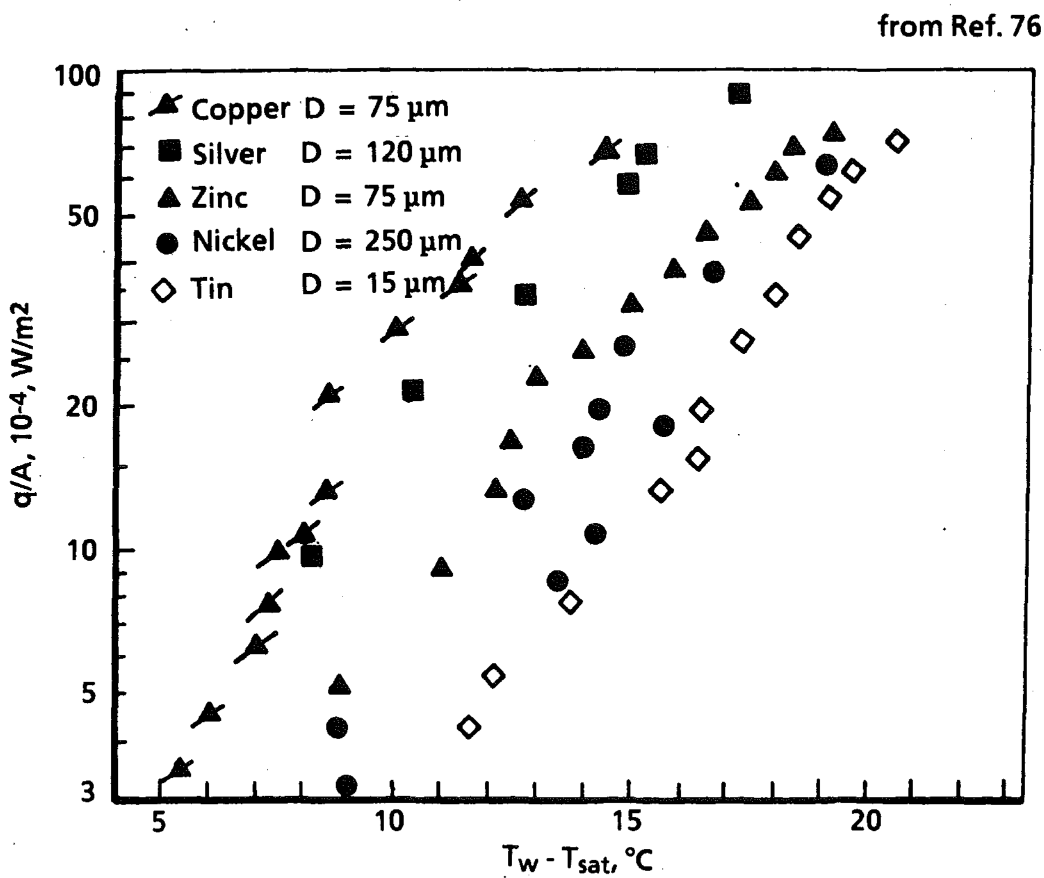


Figure 18. Effect of wall material on heat transfer.

from Ref. 78

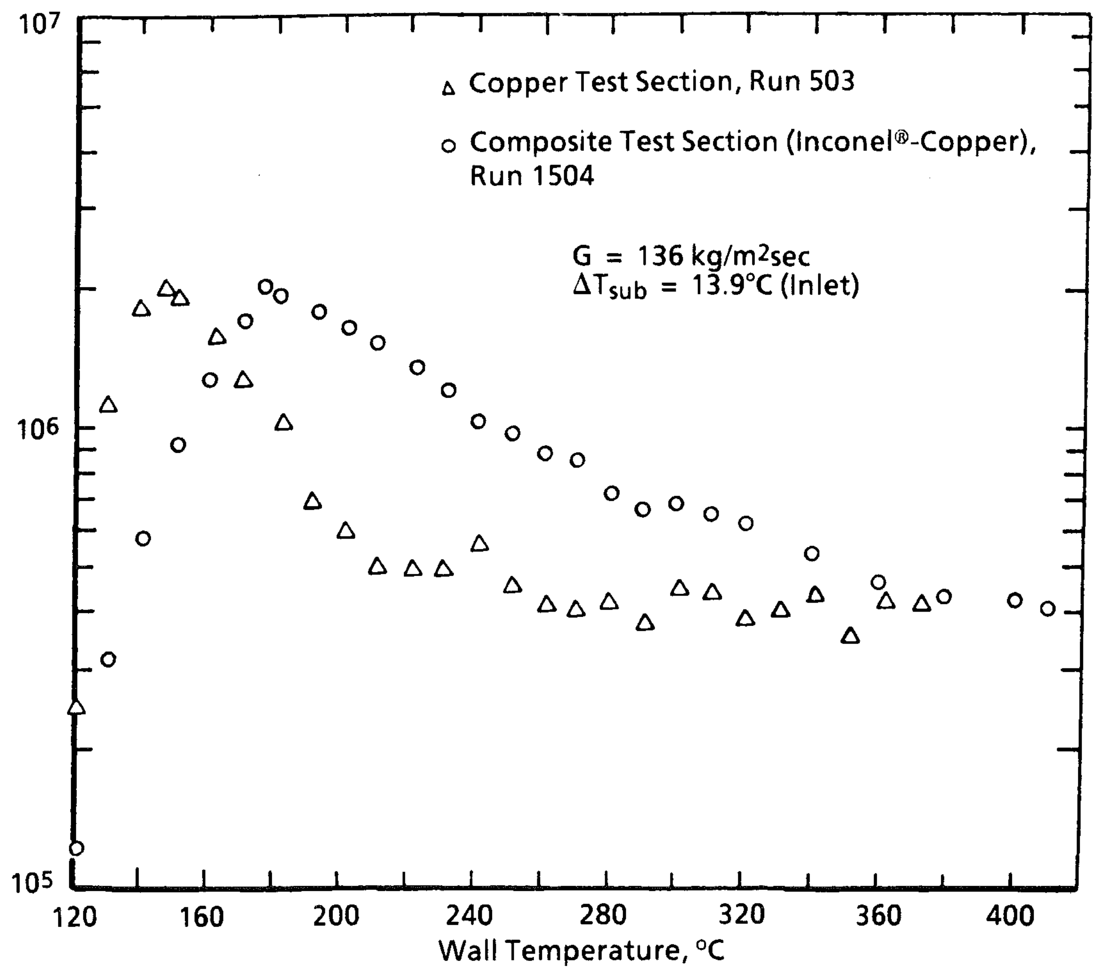


Figure 19. Effect of wall material on CHF.

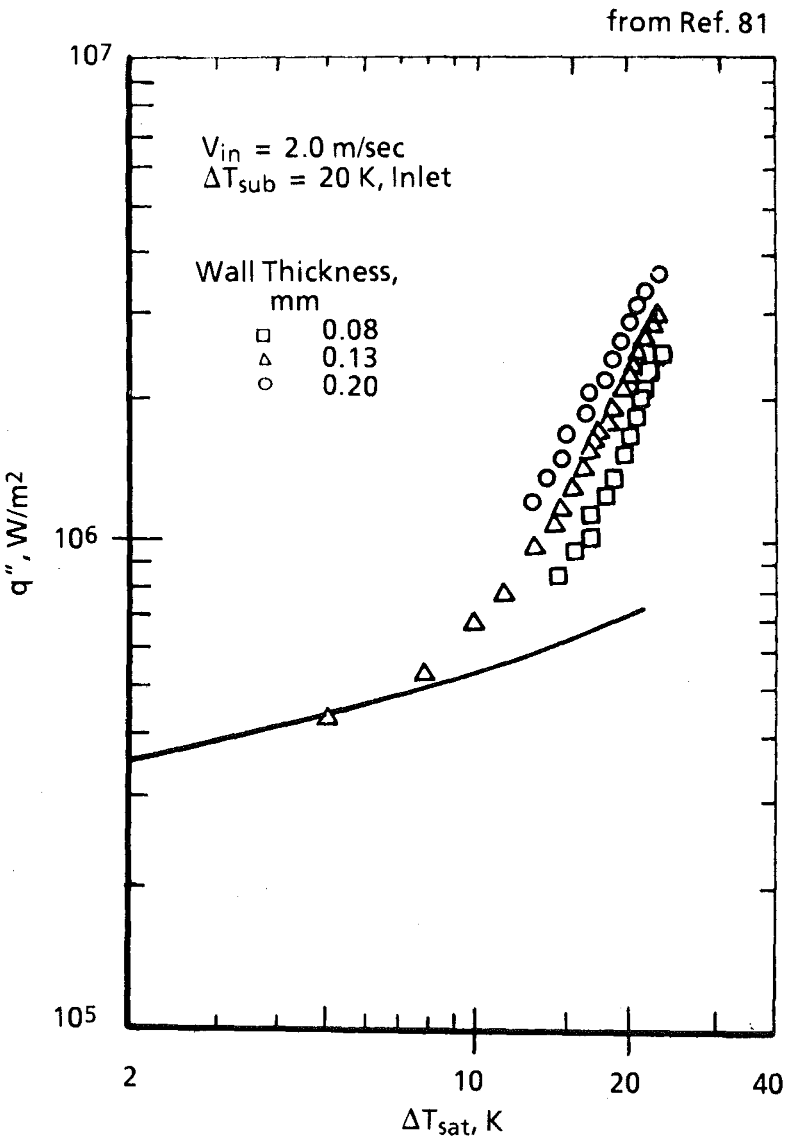


Figure 20. Effect of wall thickness on heat transfer.

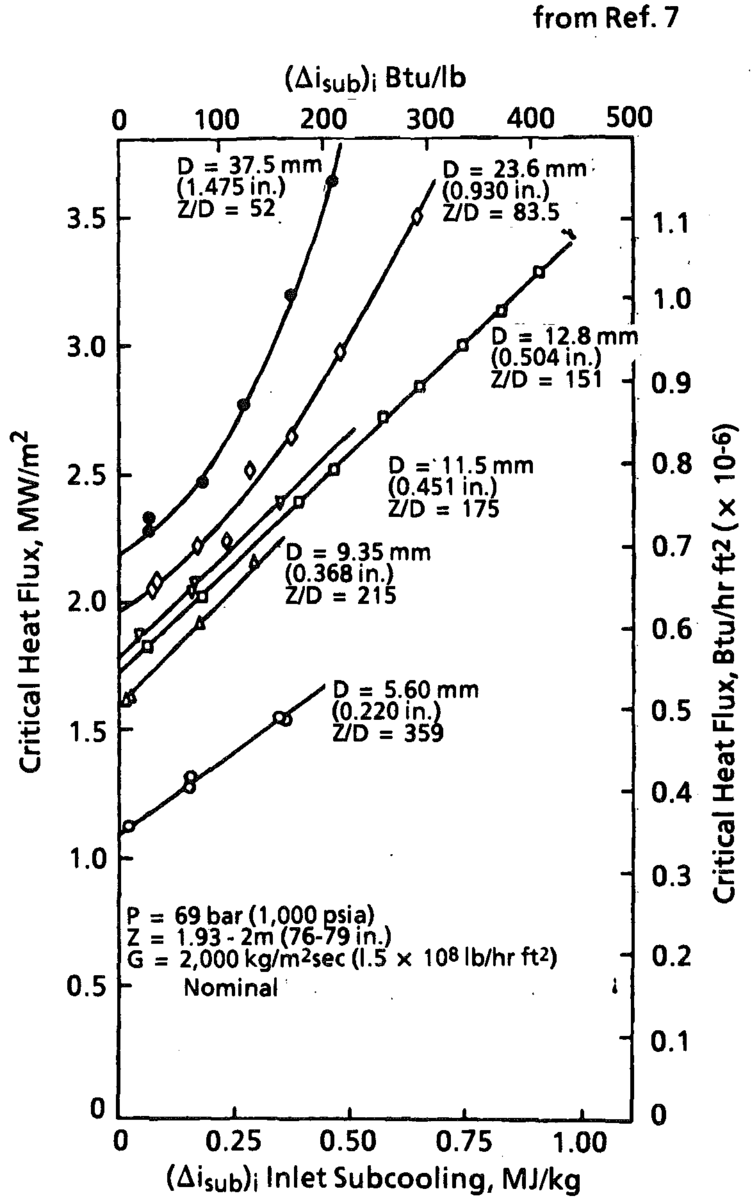


Figure 21. Influence of tube diameter on CHF at various subcoolings.

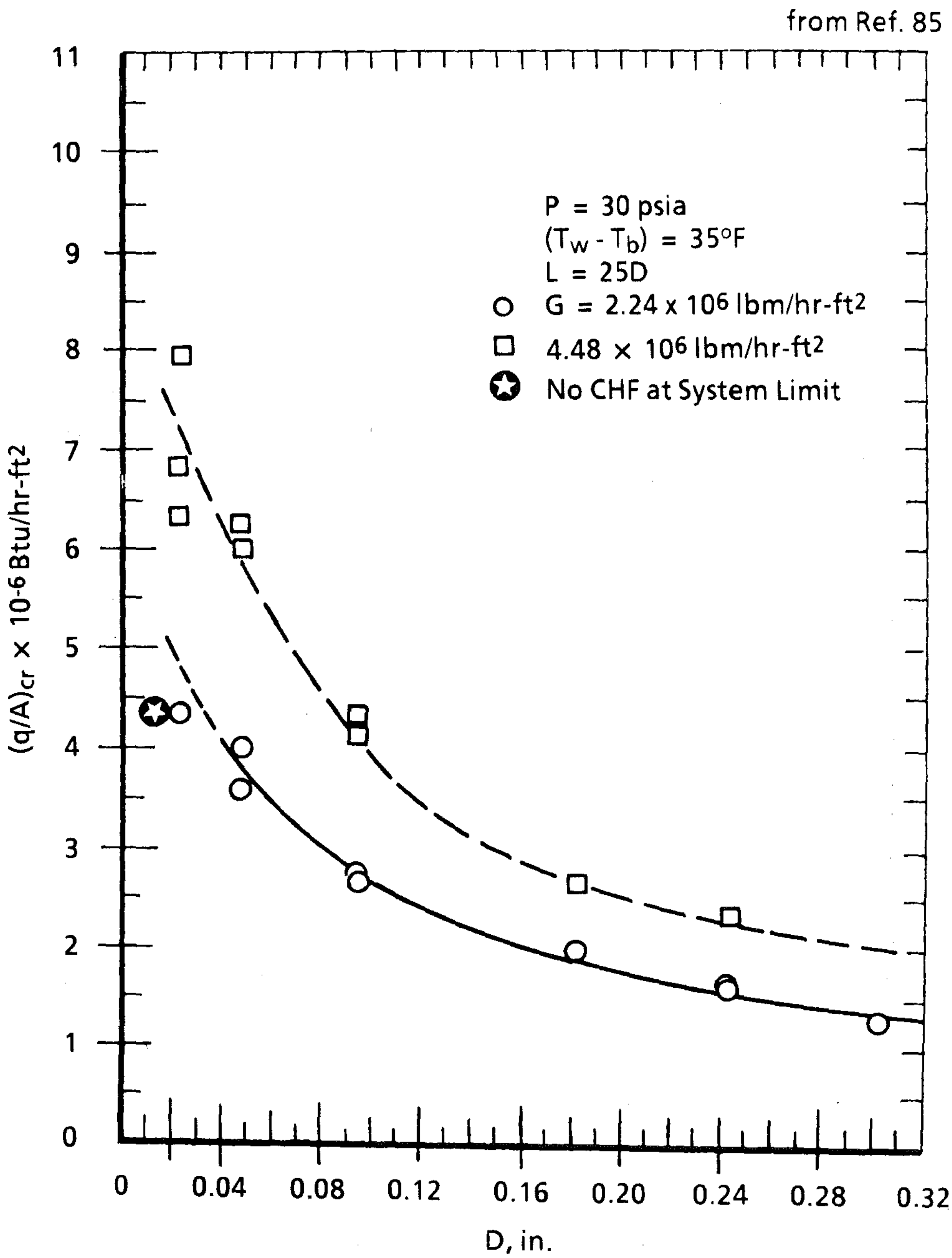


Figure 22. Effect of tube diameter on CHF.

from Ref. 85

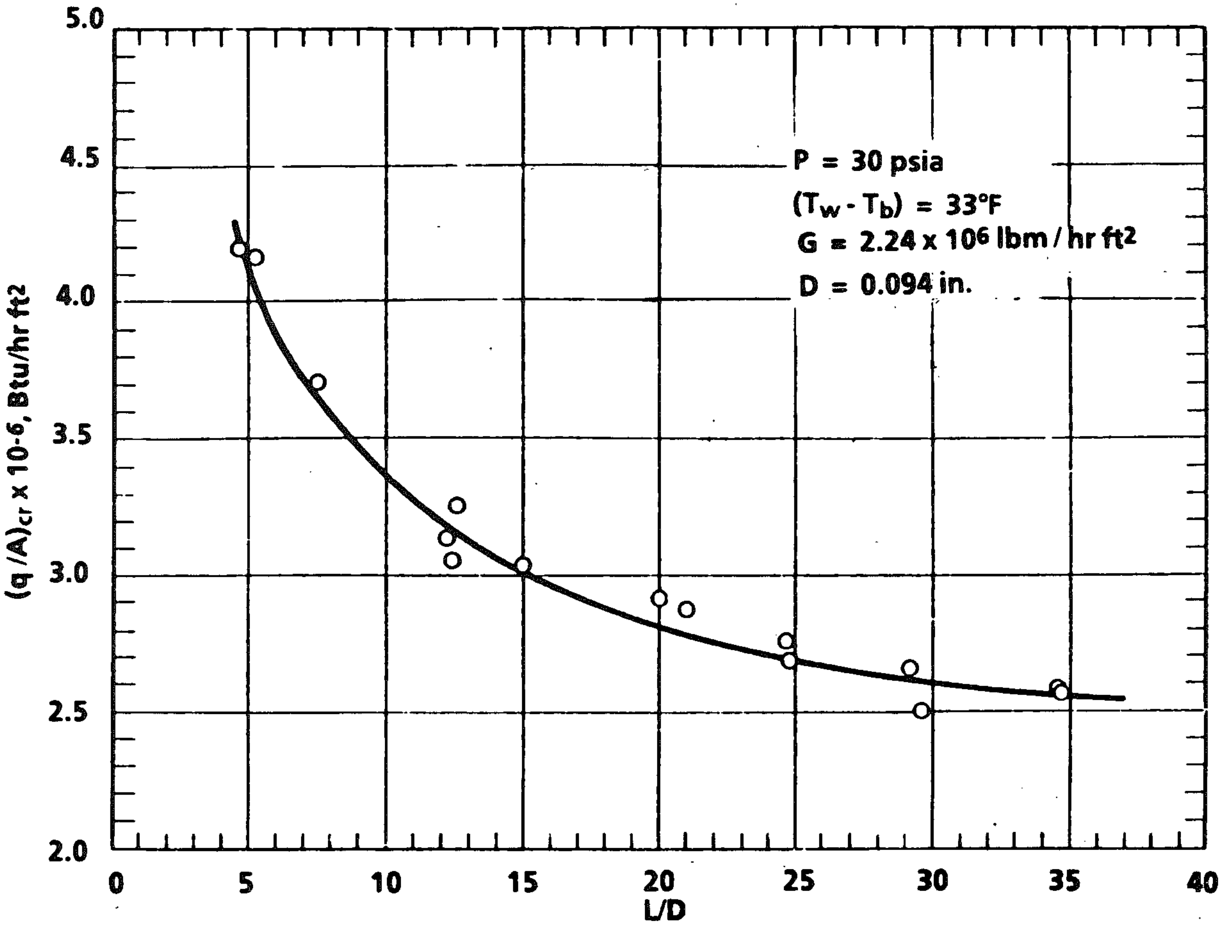


Figure 23. Effect of heated length on CHF.

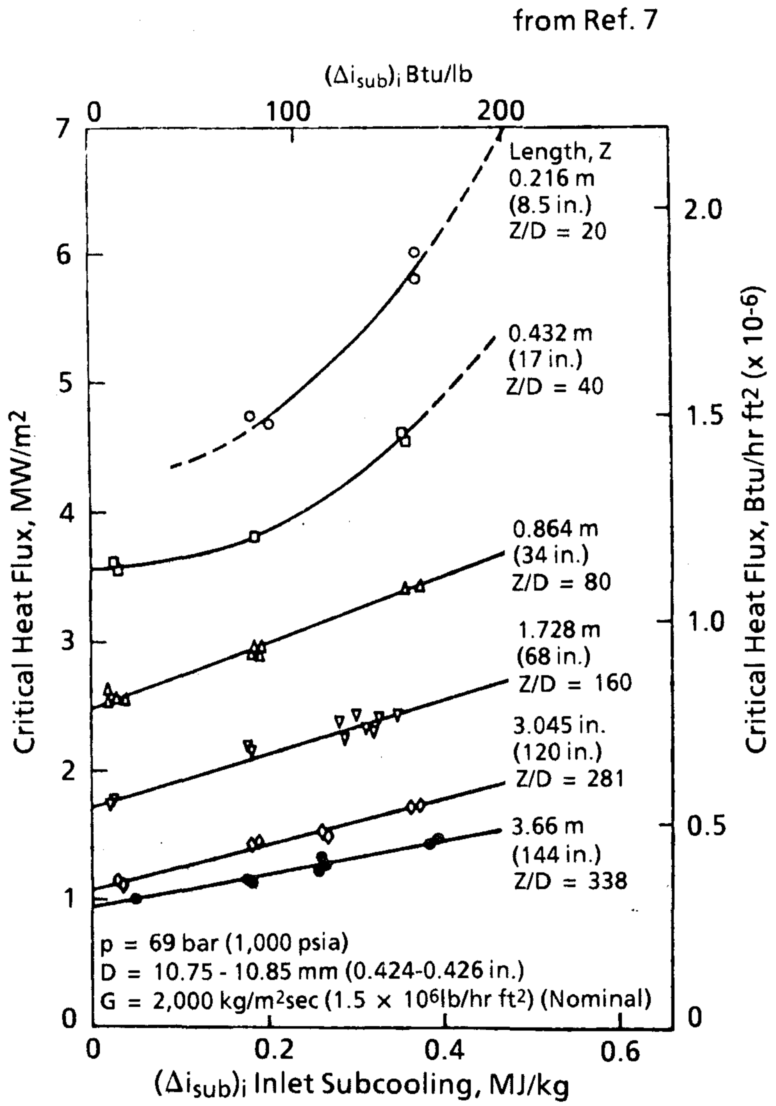


Figure 24. Effect of tube length on CHF at various subcoolings.

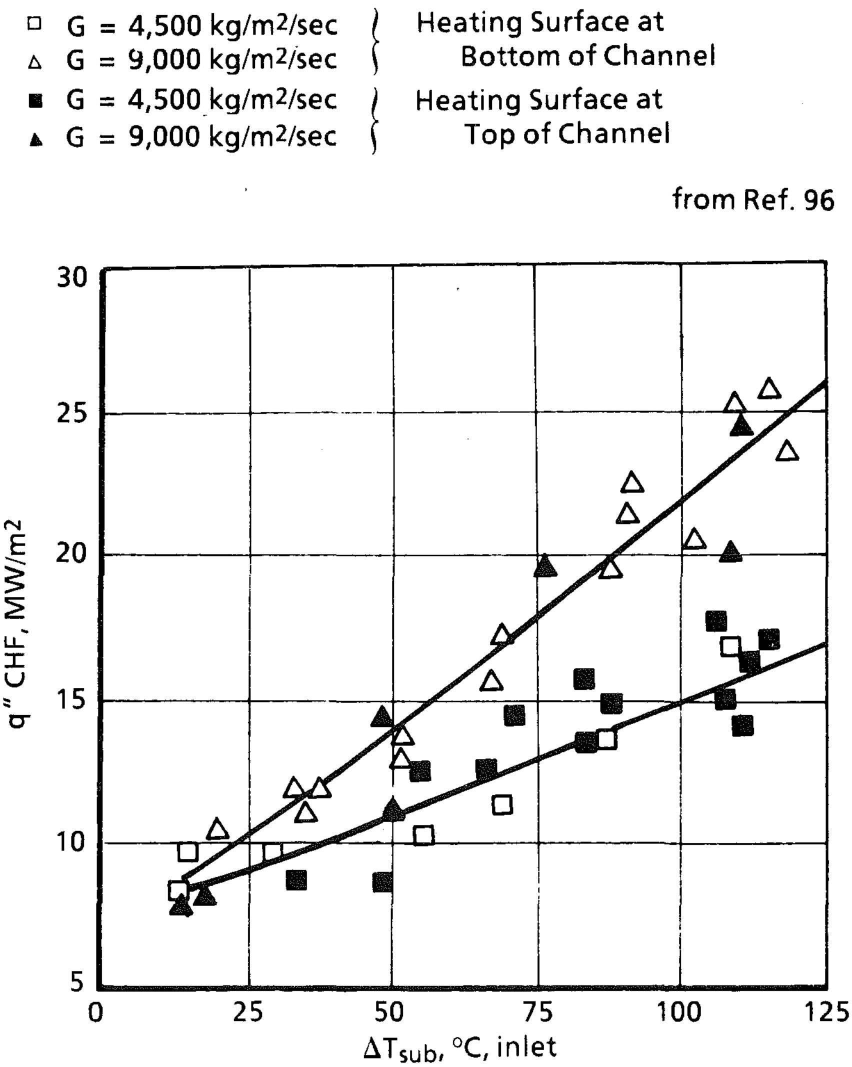


Figure 25. Effect of position of the heating surface on CHF.

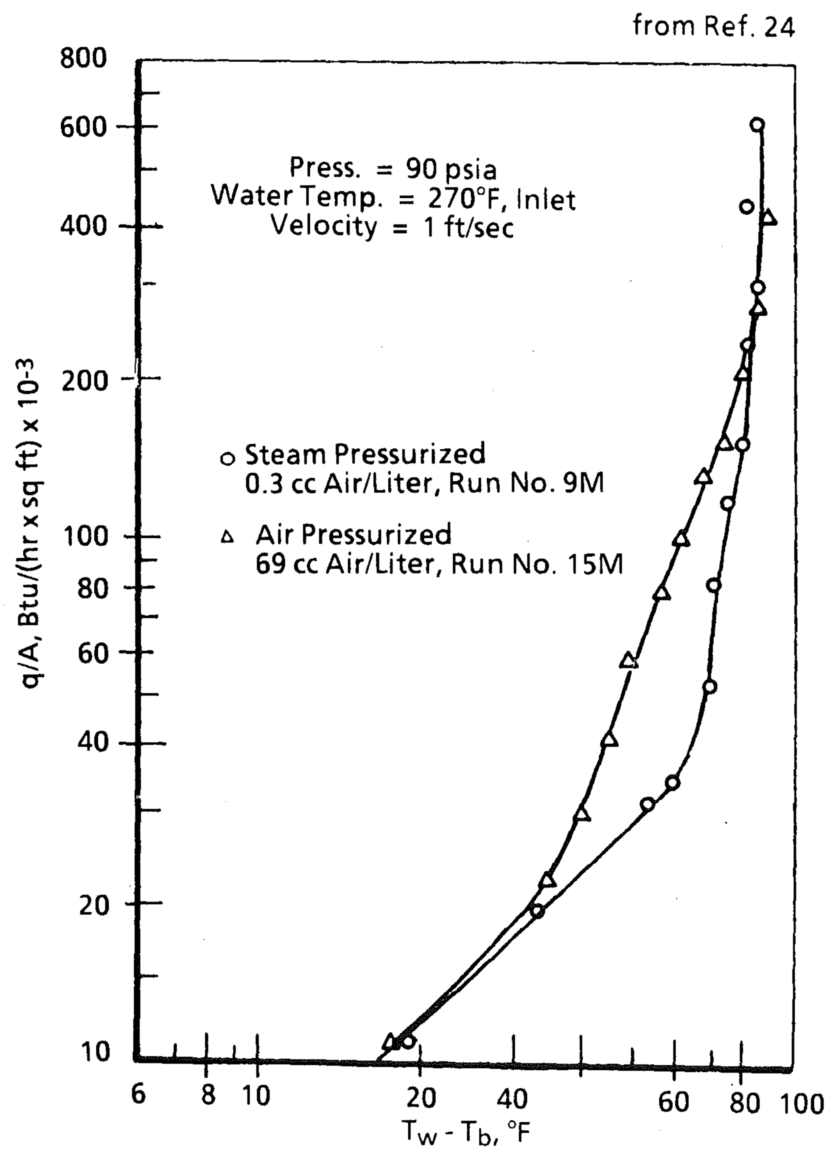


Figure 26. Effect of gas content on heat transfer.

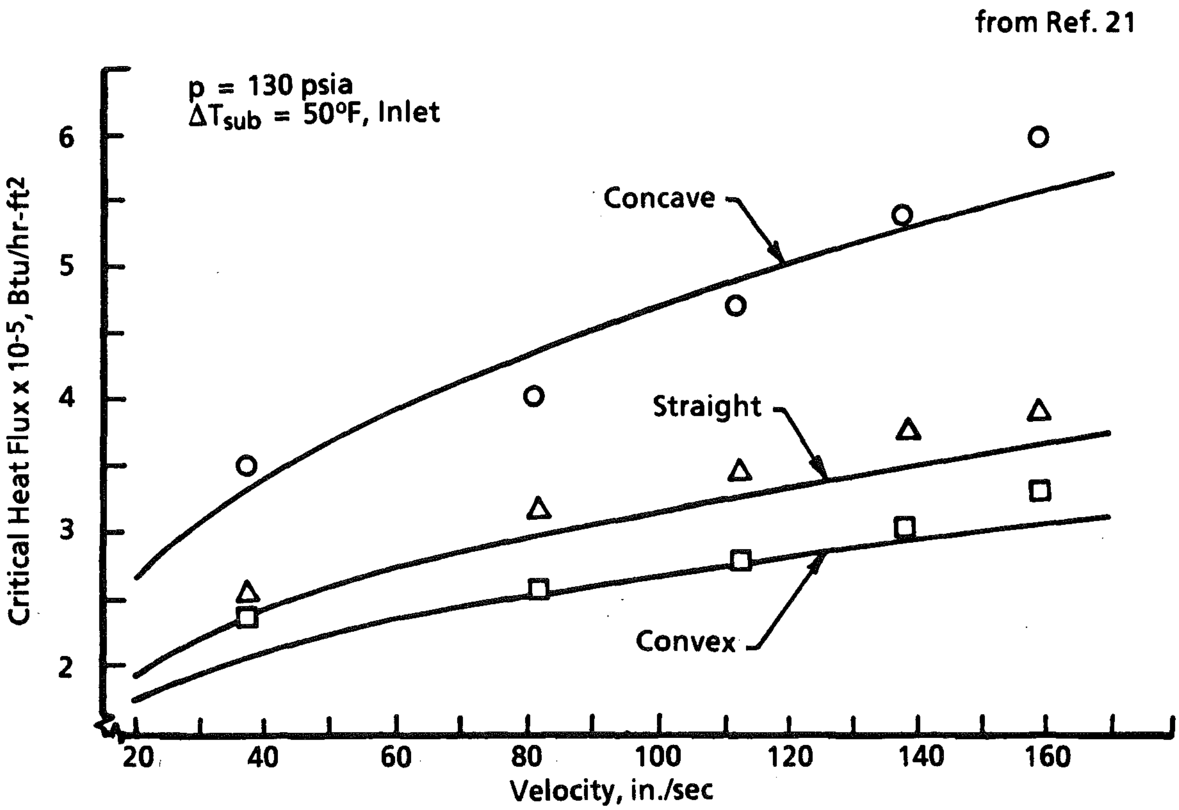


Figure 27. Effect of curved test section on CHF.

from Ref. 111

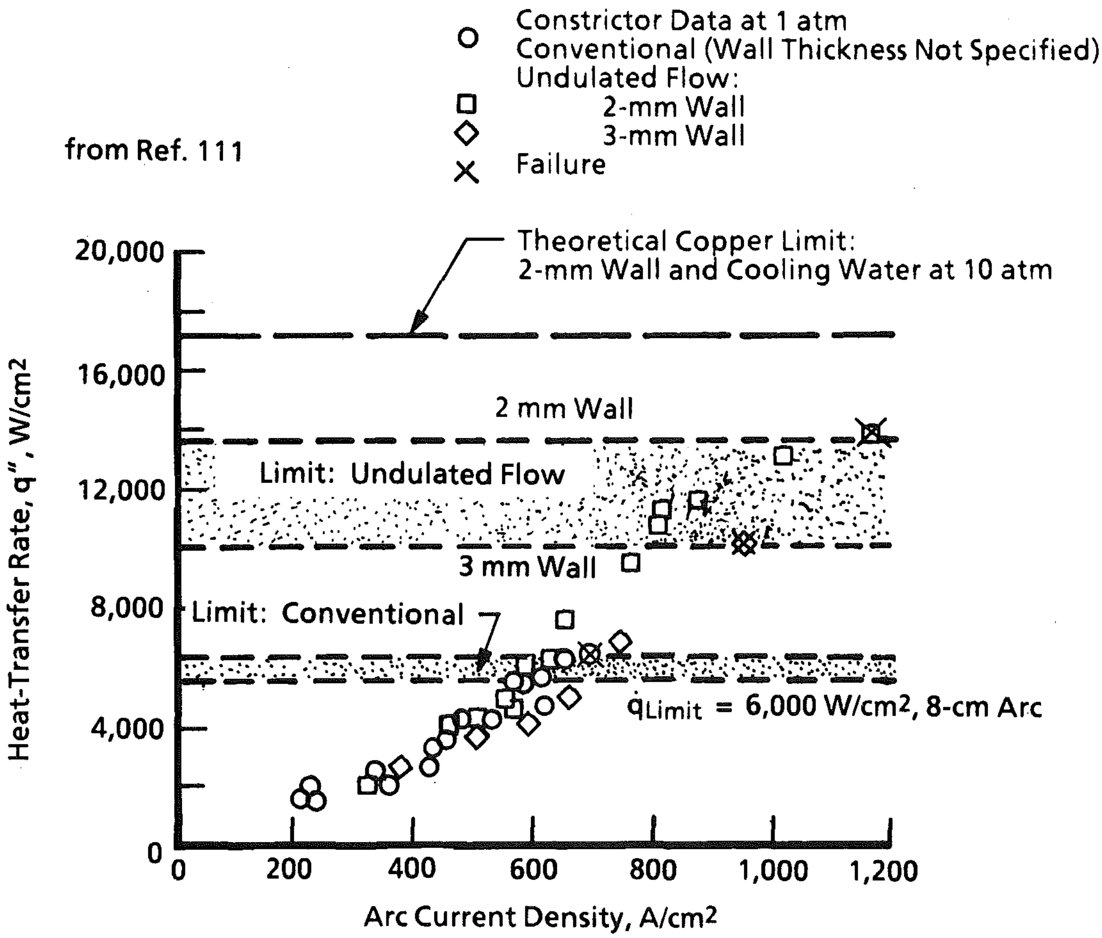
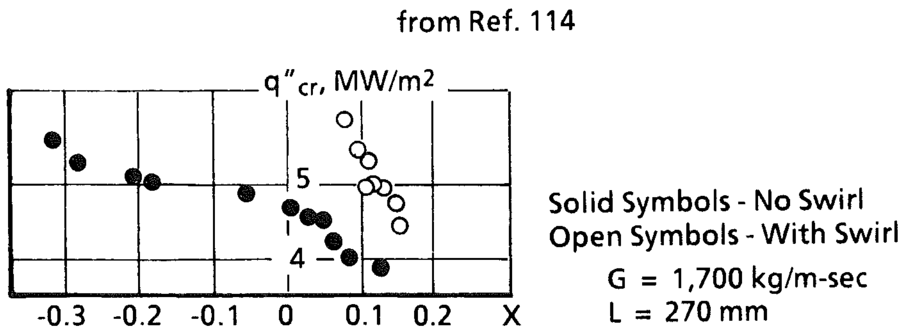
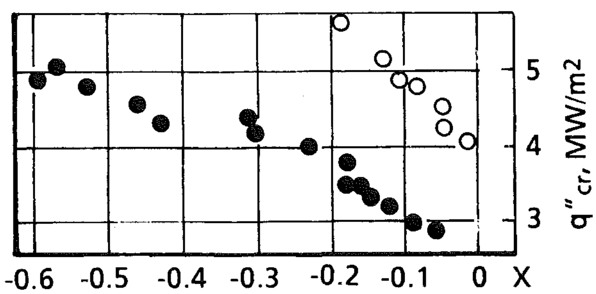


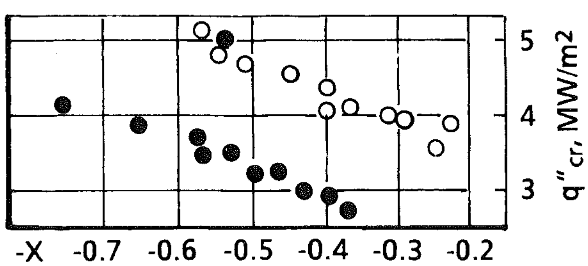
Figure 28. Effect of undulating flow on heat transfer.



a. p = 98.1 bar

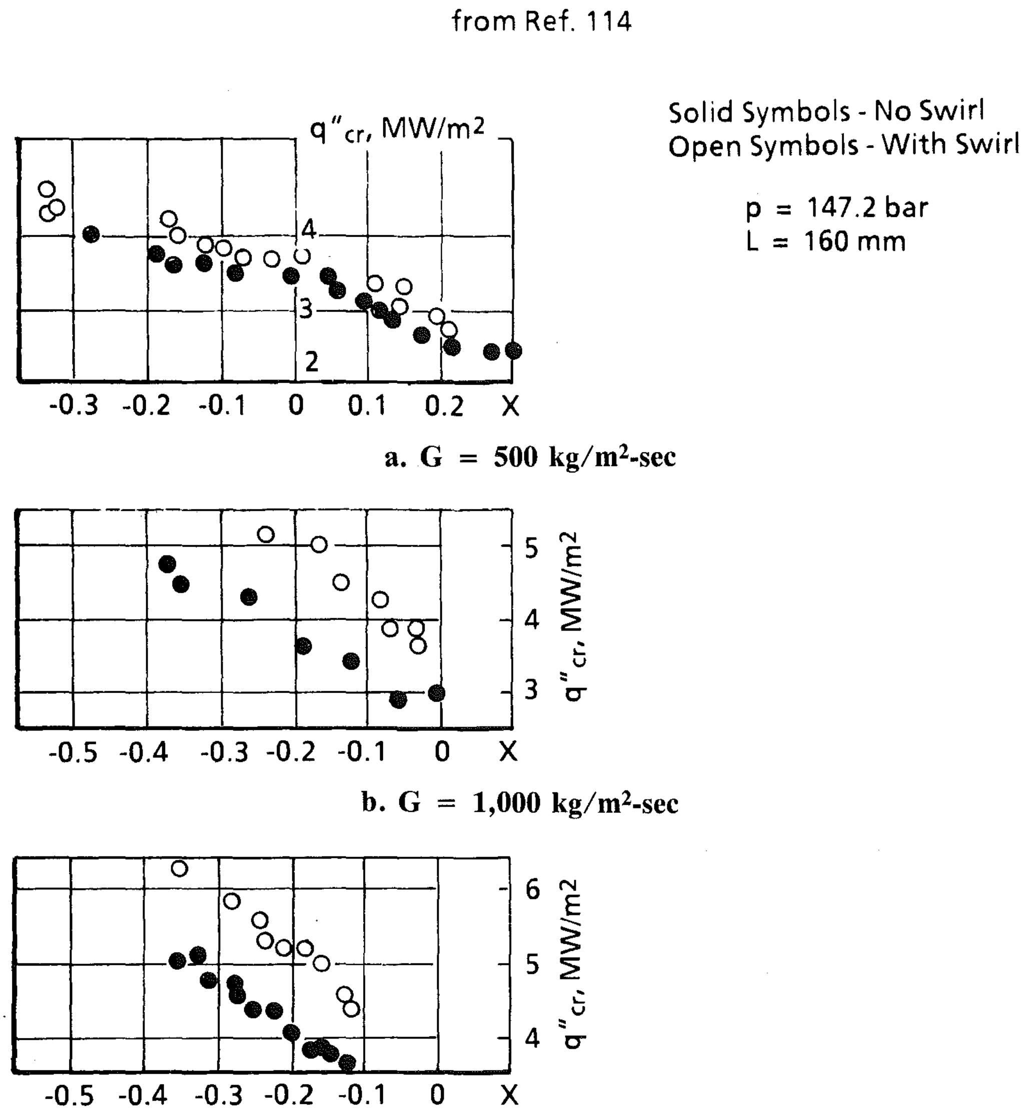


b. p = 147.2 bar



c. p = 186.5 bar

Figure 29. Pressure effect with flow swirl.



a. G = 500 kg/m²-sec

b. G = 1,000 kg/m²-sec

c. G = 1,700 kg/m²-sec

Figure 30. Mass velocity effect with flow swirl.

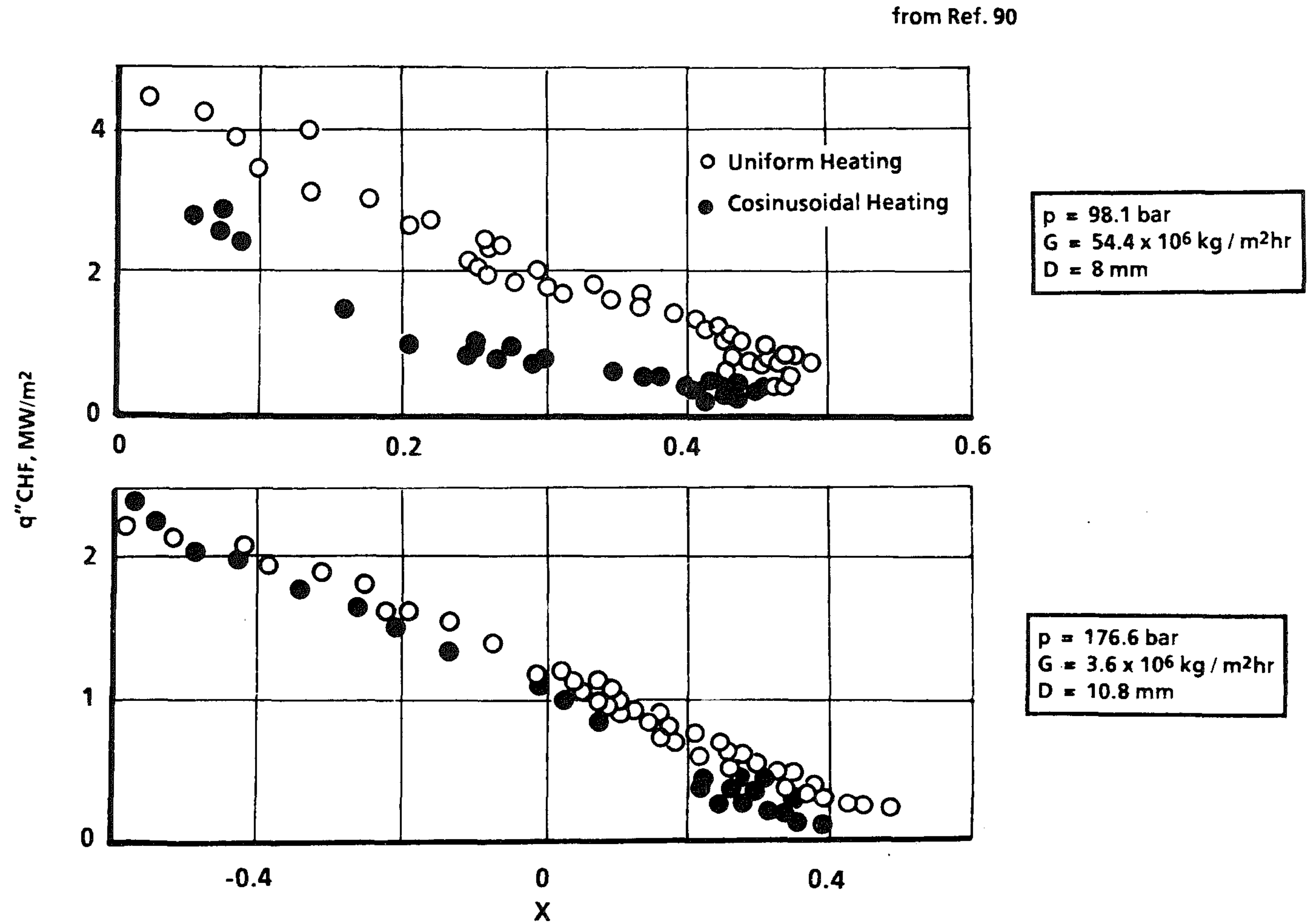


Figure 31. Effect of nonuniform heating on CHF.

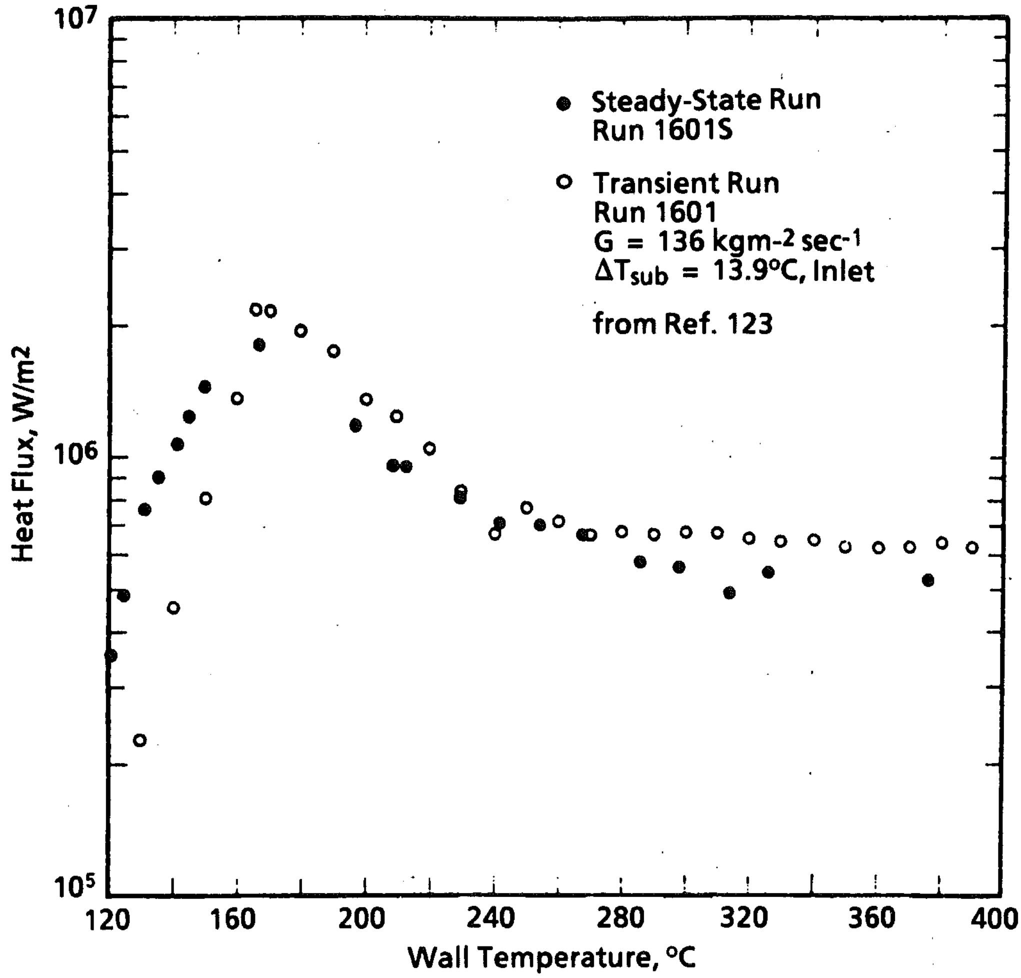
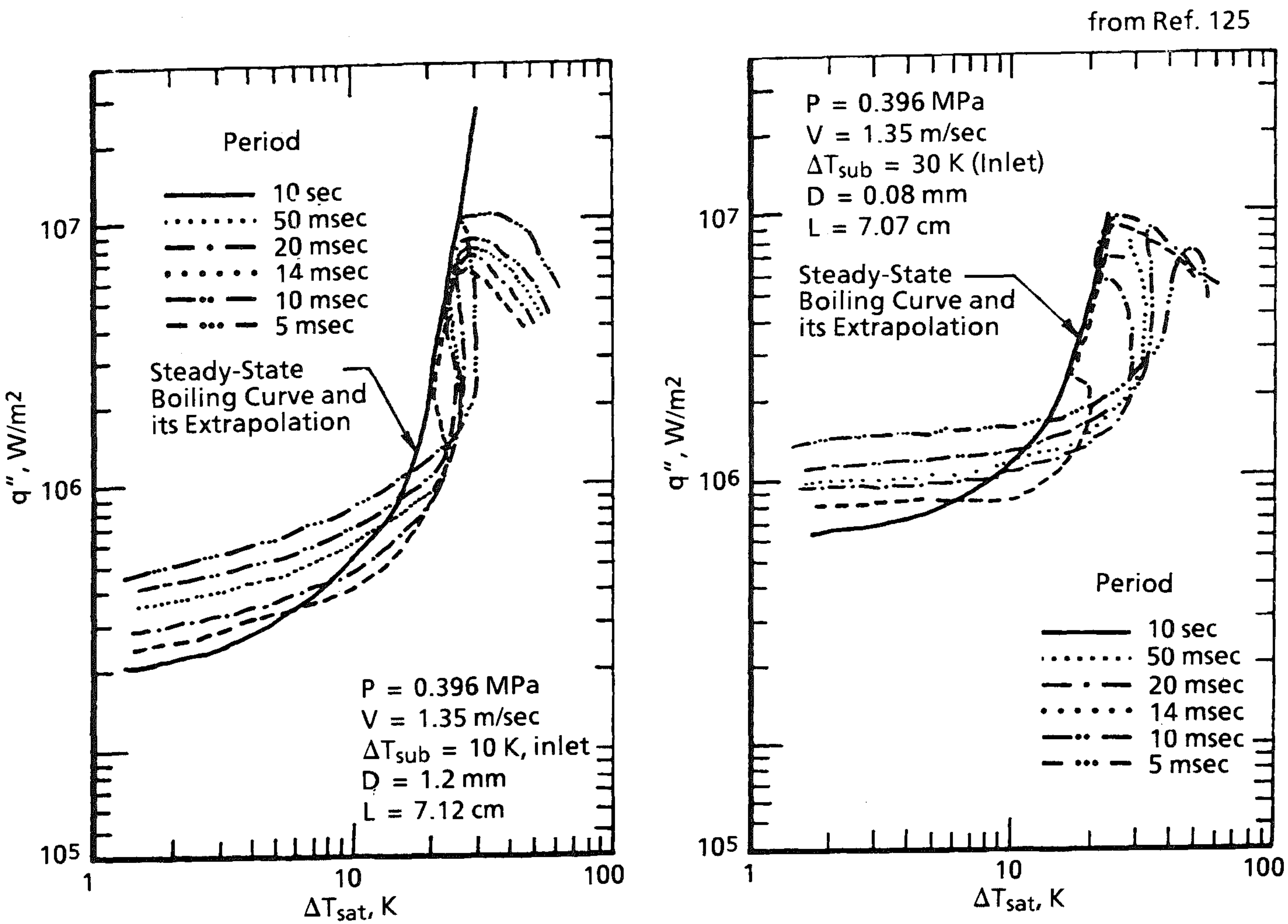


Figure 32. Comparison of steady-state and transient heating data.



a. D = 1.2 mm and T_{sub} = 10 K, inlet   b. D = 0.8 mm and ΔT_{sub} = 30 K, inlet

Figure 33. Effect of transient heating on boiling curve.

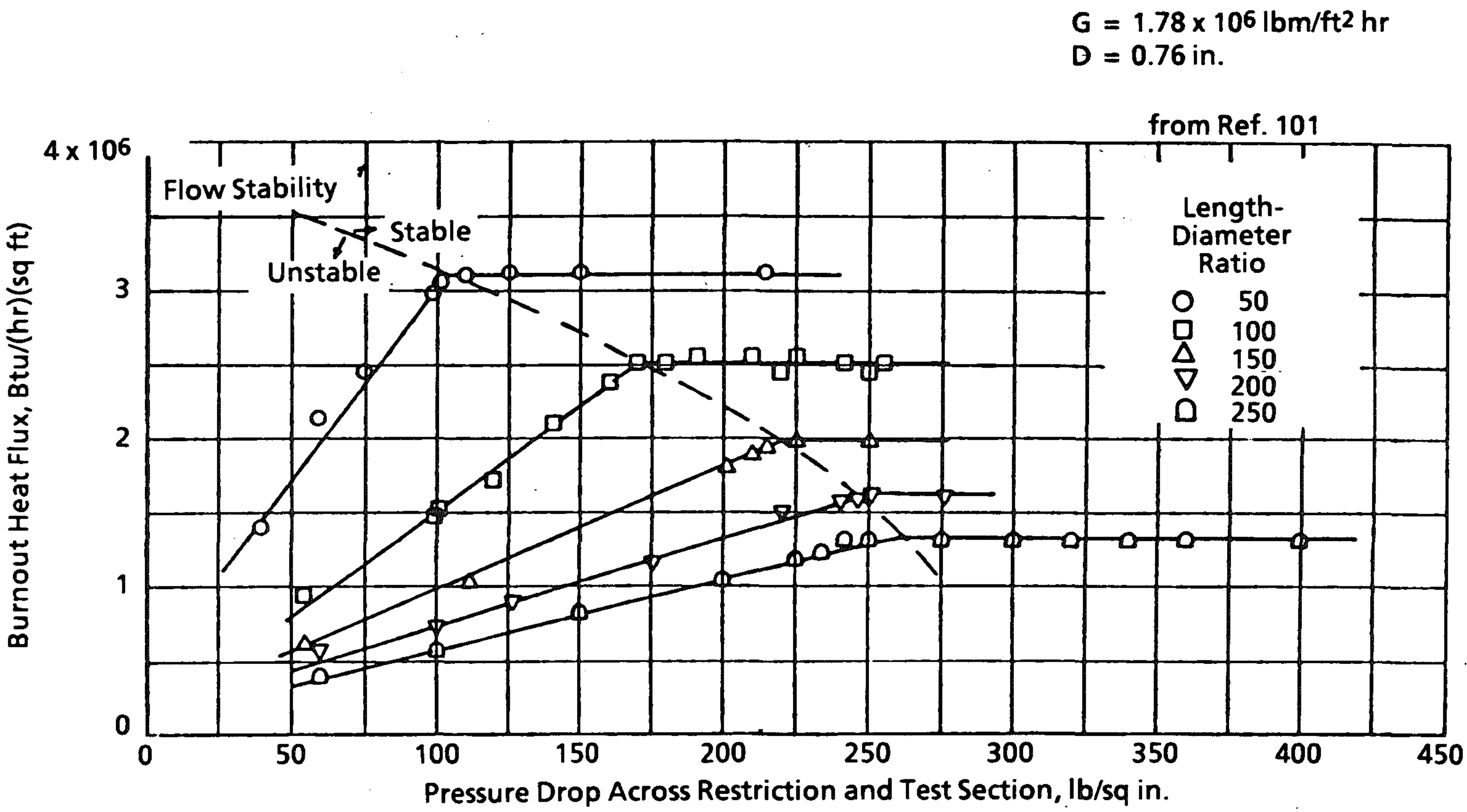


Figure 34. Effect of flow restriction on flow stability and burnout.

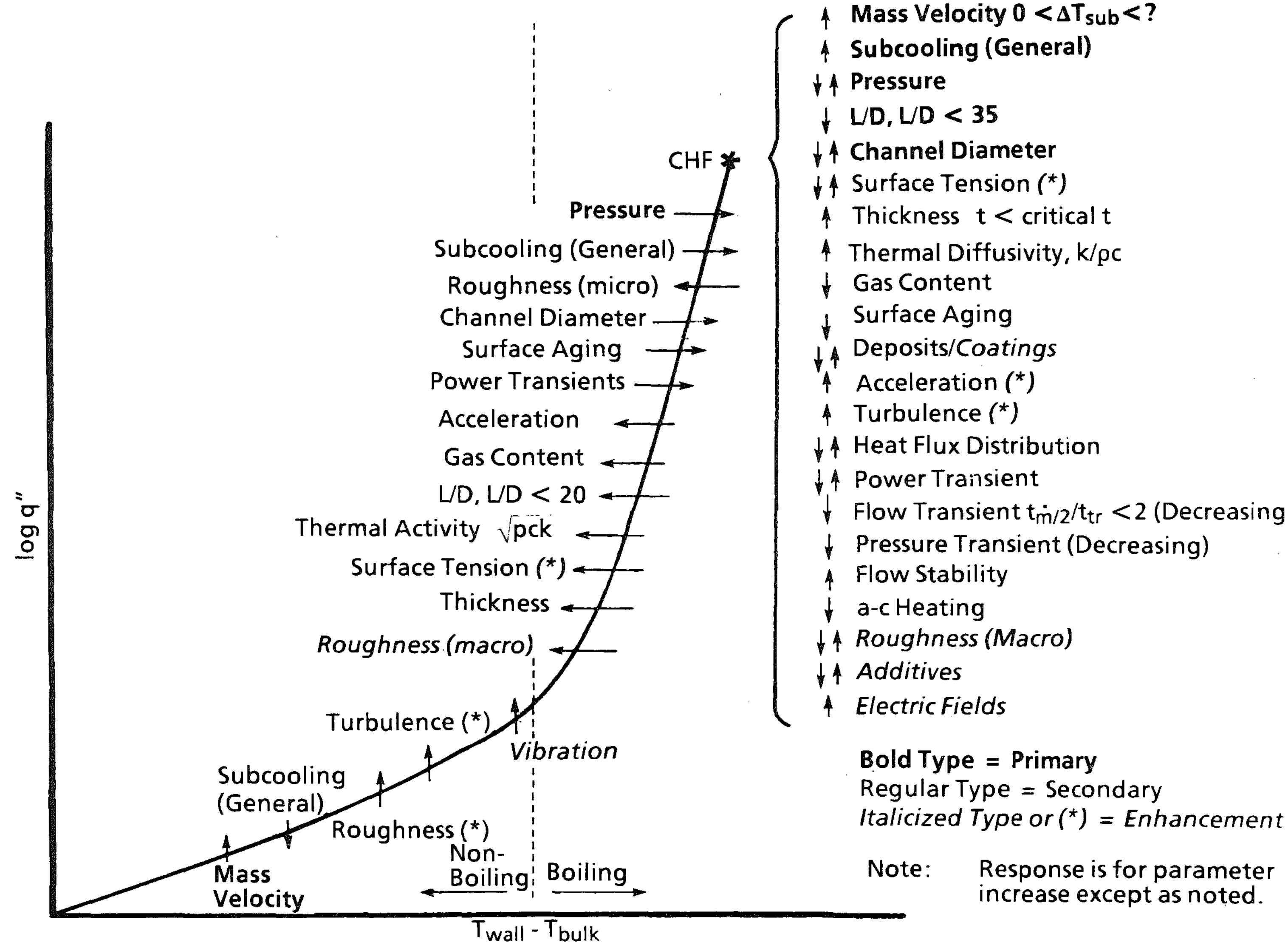


Figure 35. Parametric trends for subcooled, forced-convection cooling.

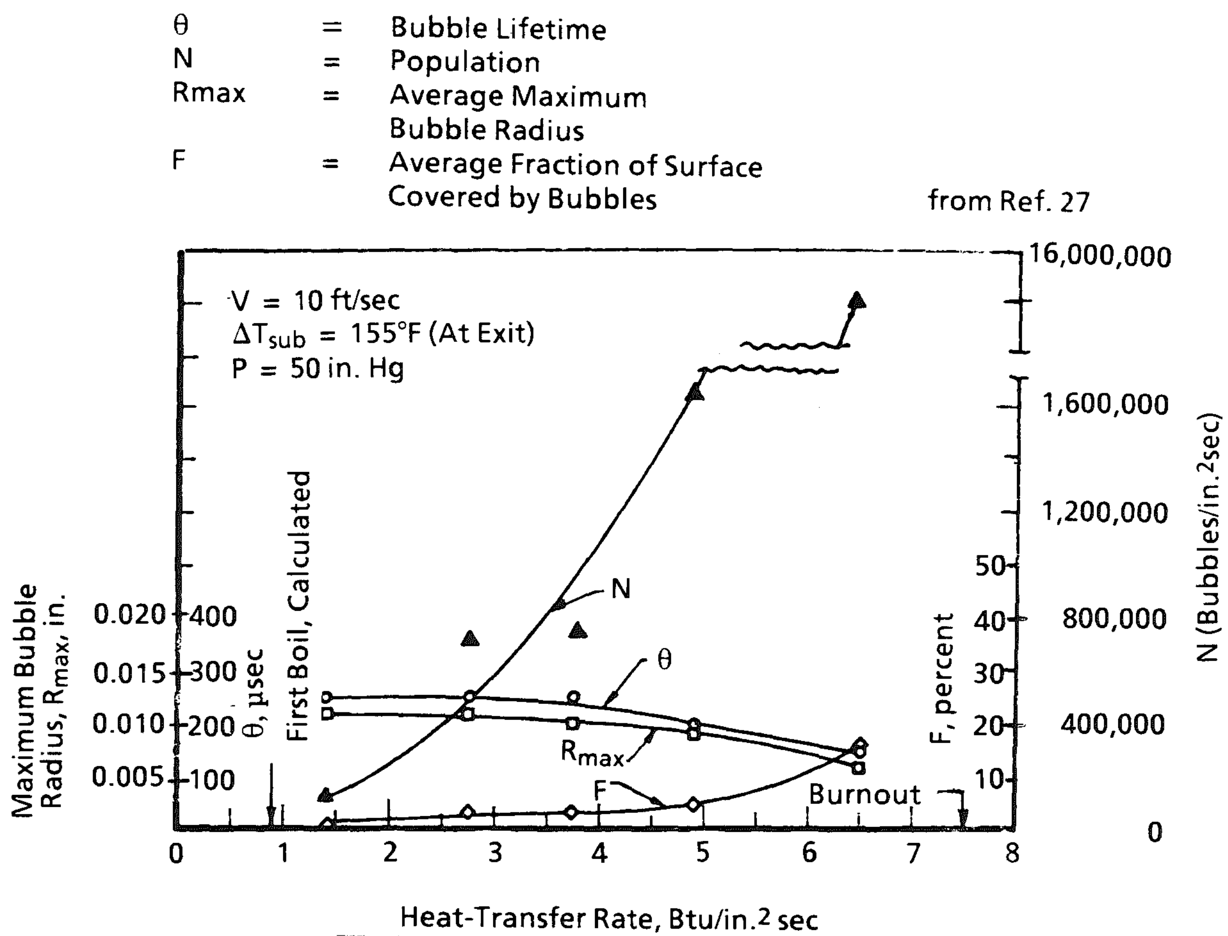


Figure 36. Effect of heat-transfer rate on bubble characteristics.

from Ref. 161

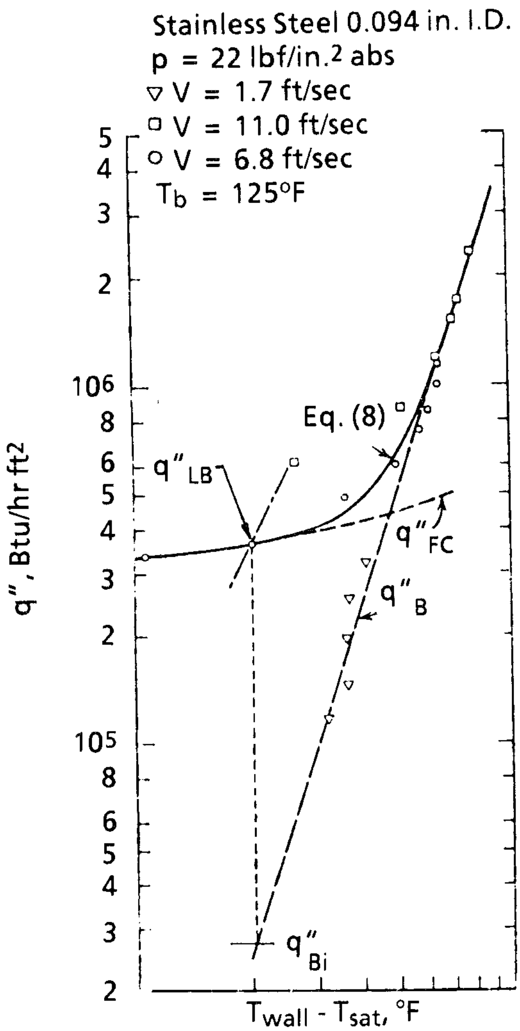


Figure 37. Construction of the partial boiling curve.

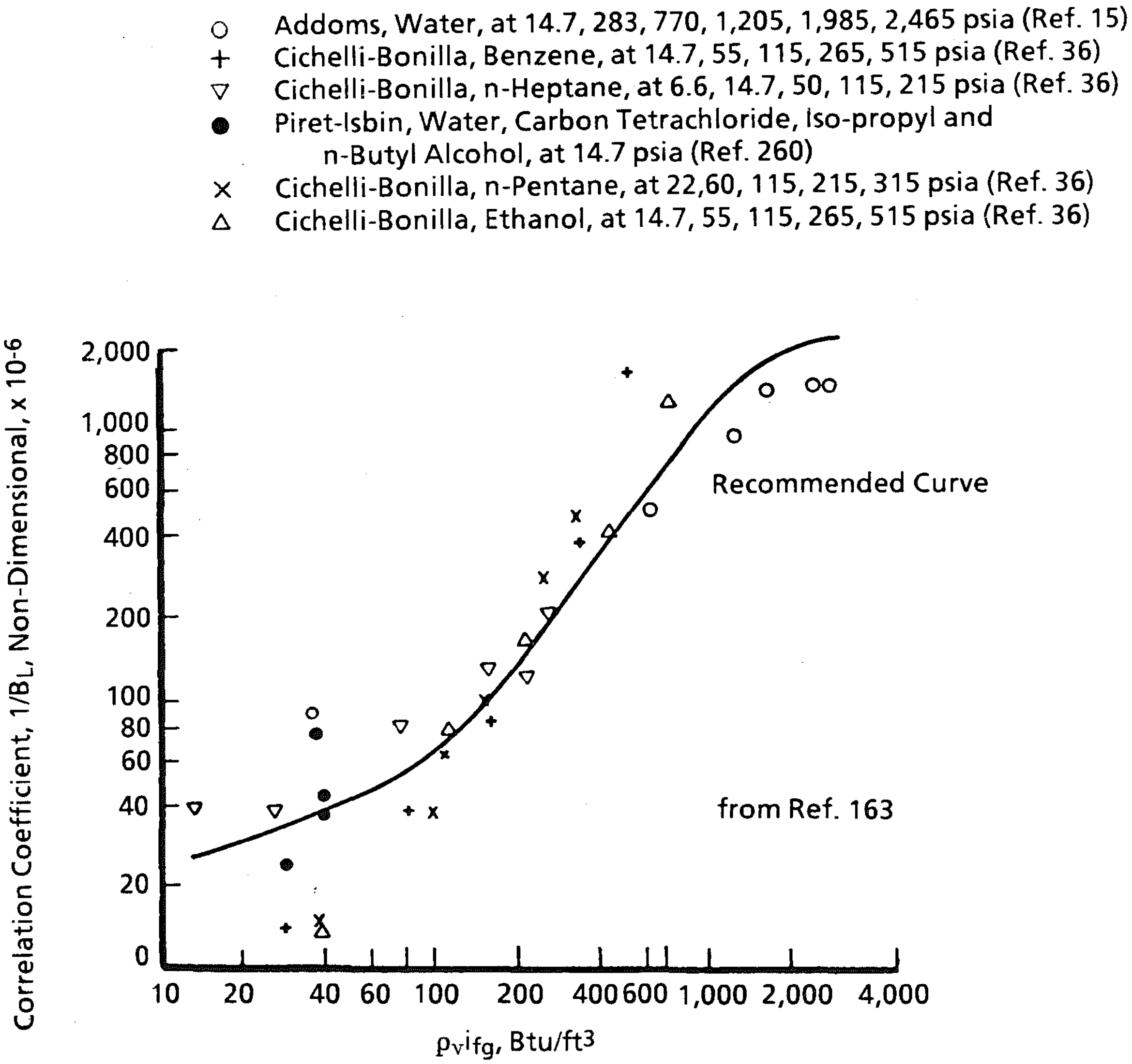


Figure 38. Determination of the coefficient B_L in the Levy correlation.

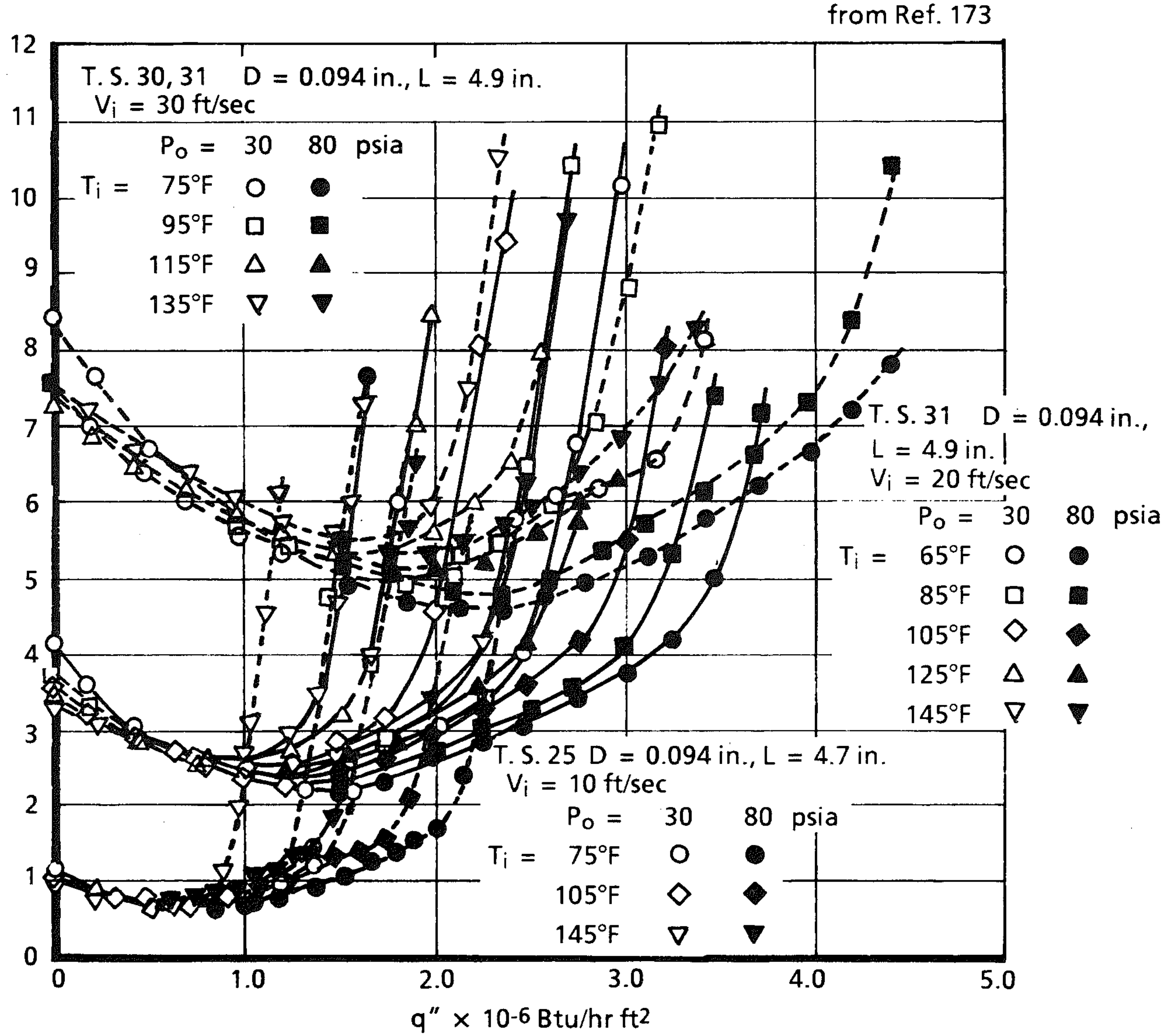


Figure 39. Dependence of overall pressure drop on operating conditions.

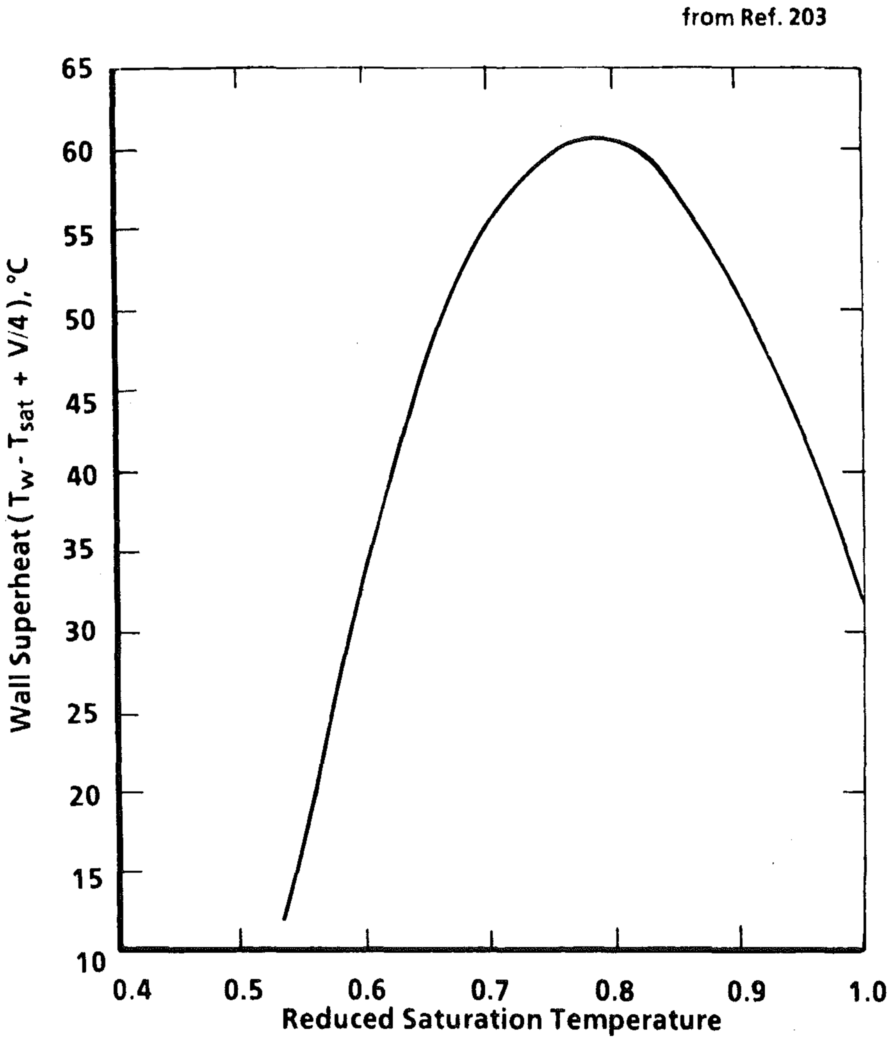


Figure 40. Generalized wall superheat at burnout.

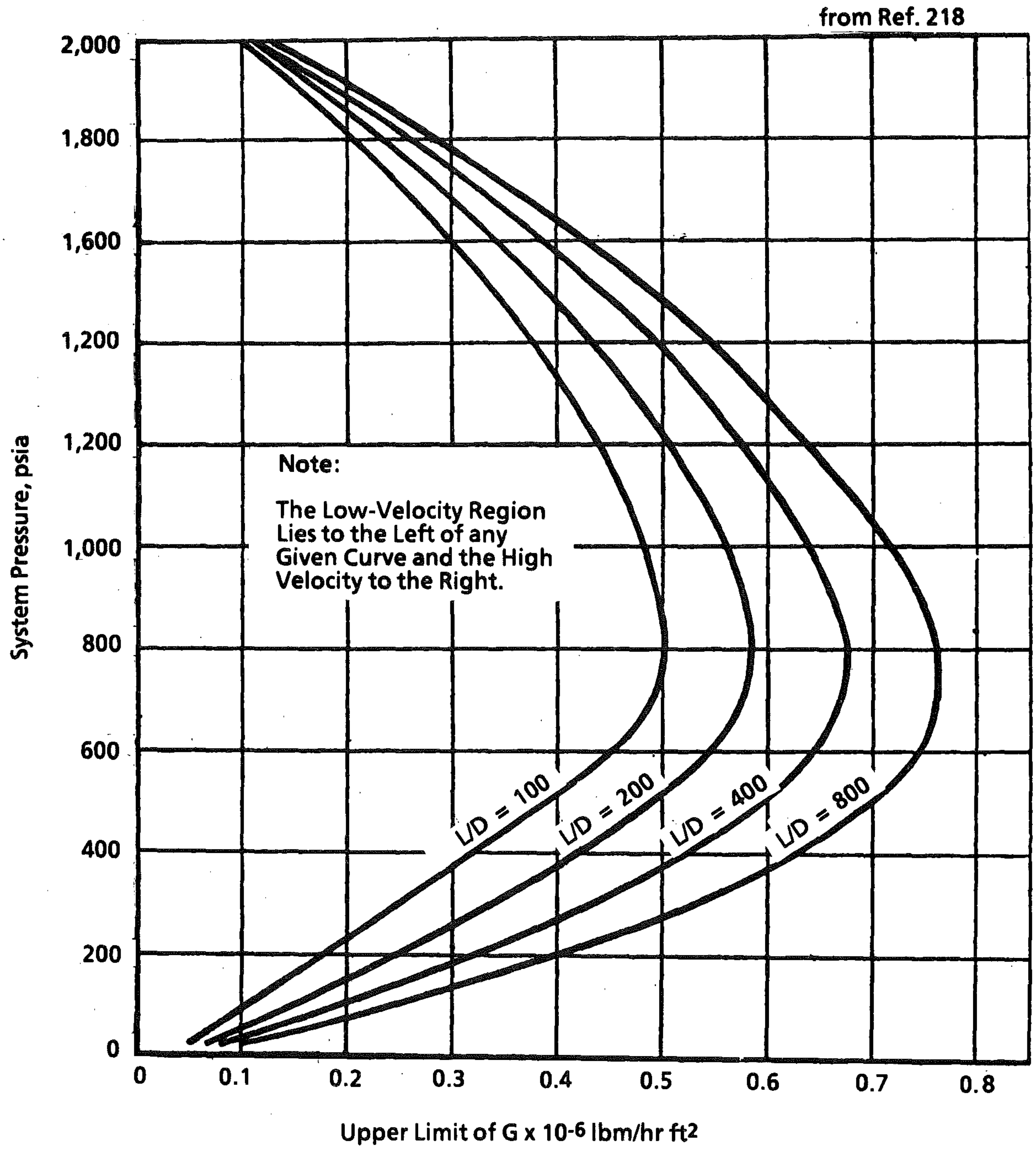
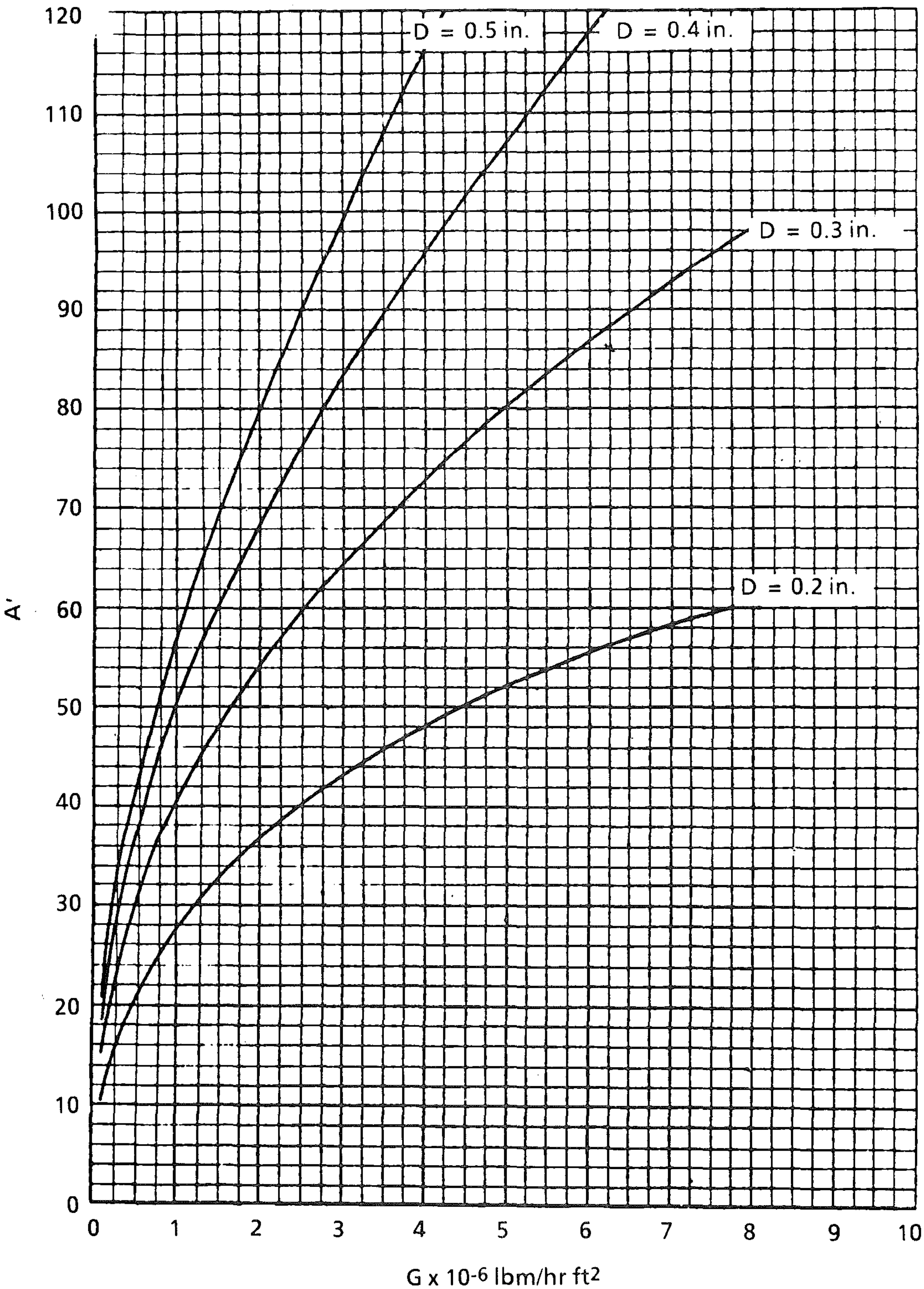
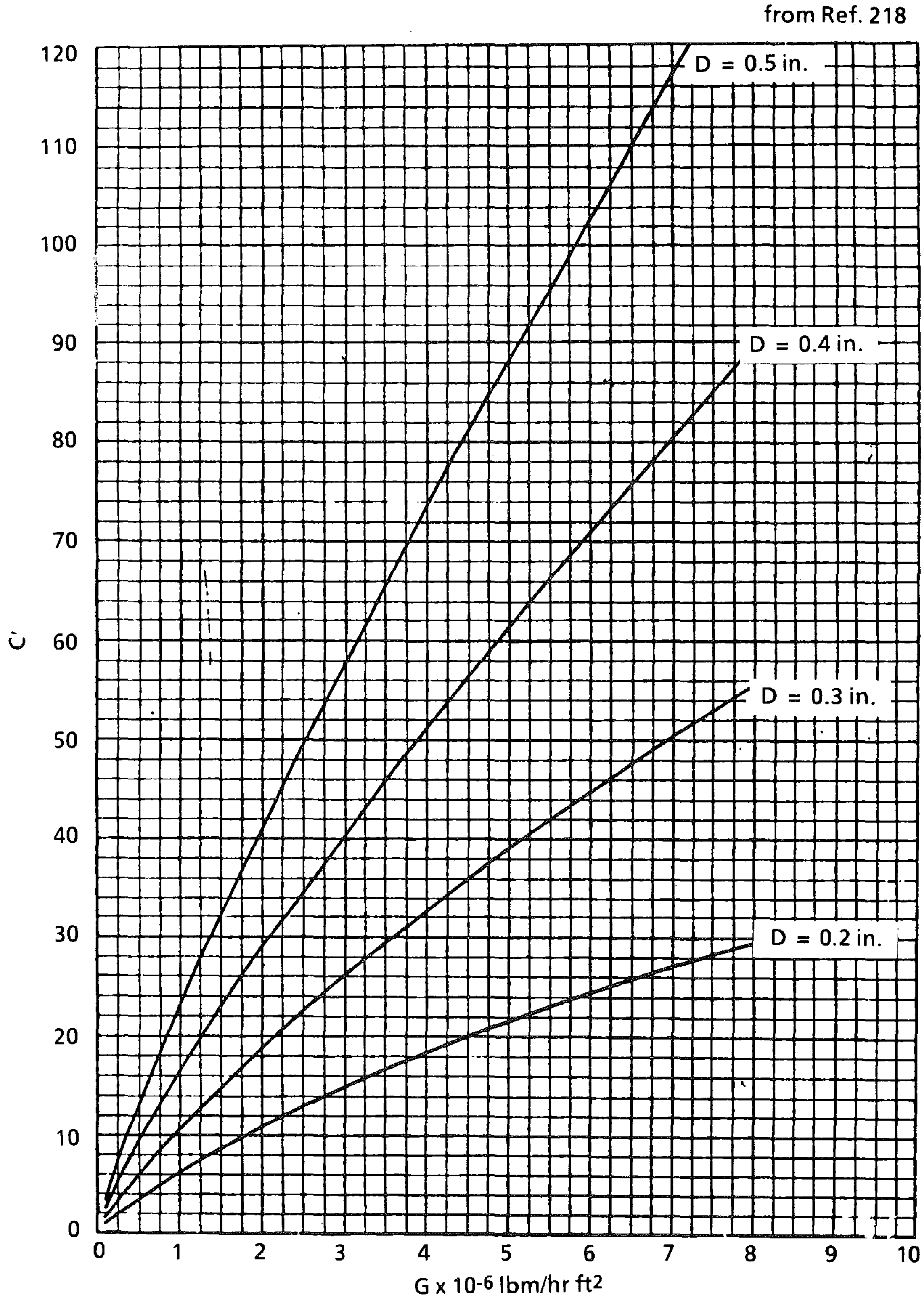


Figure 41. Approximate limits of velocity regimes for round tubes.

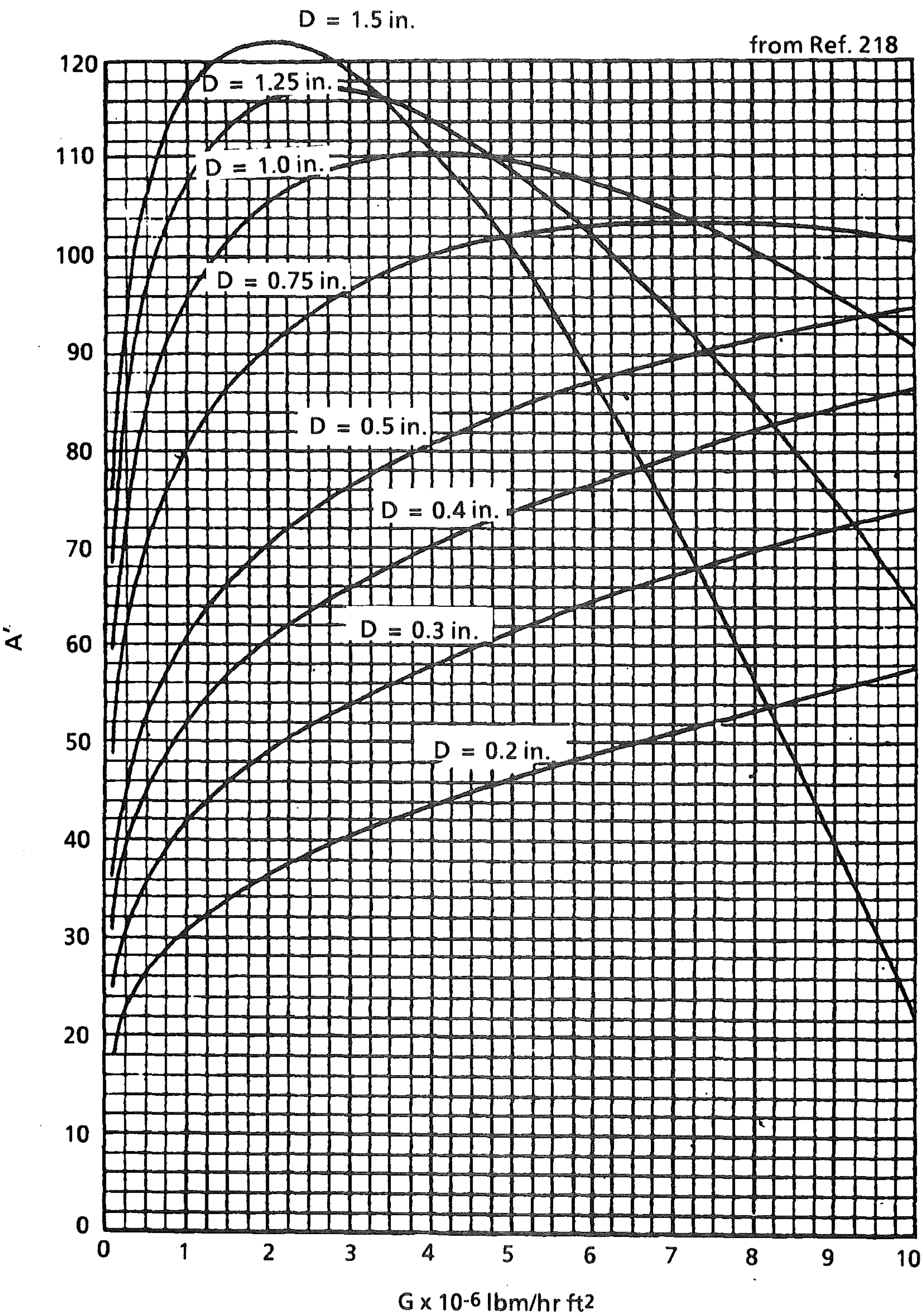
from Ref. 218



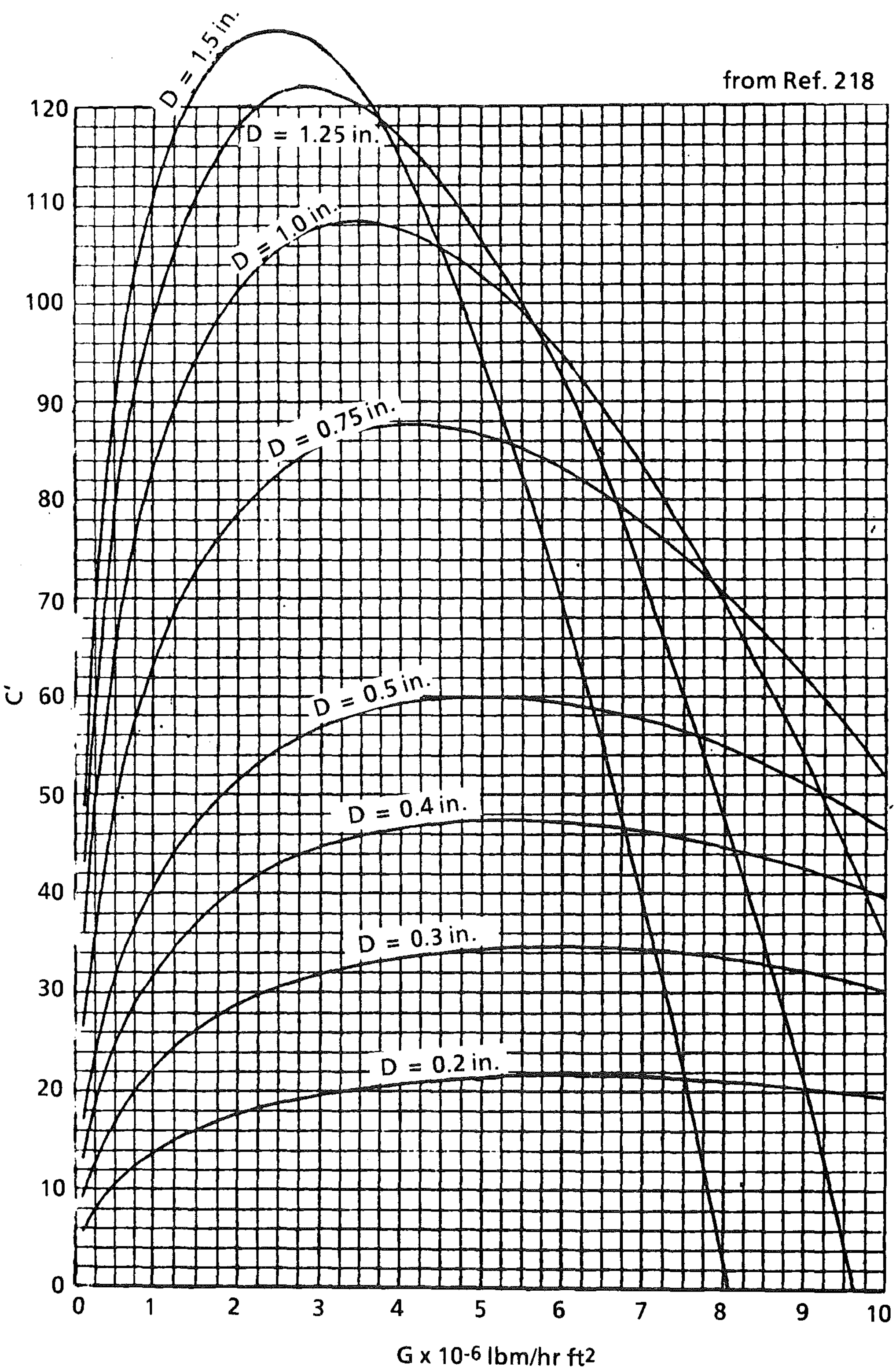
a. A′ for p = 560 psia
Figure 42. Constant inputs for the Thompson and Macbeth correlation.



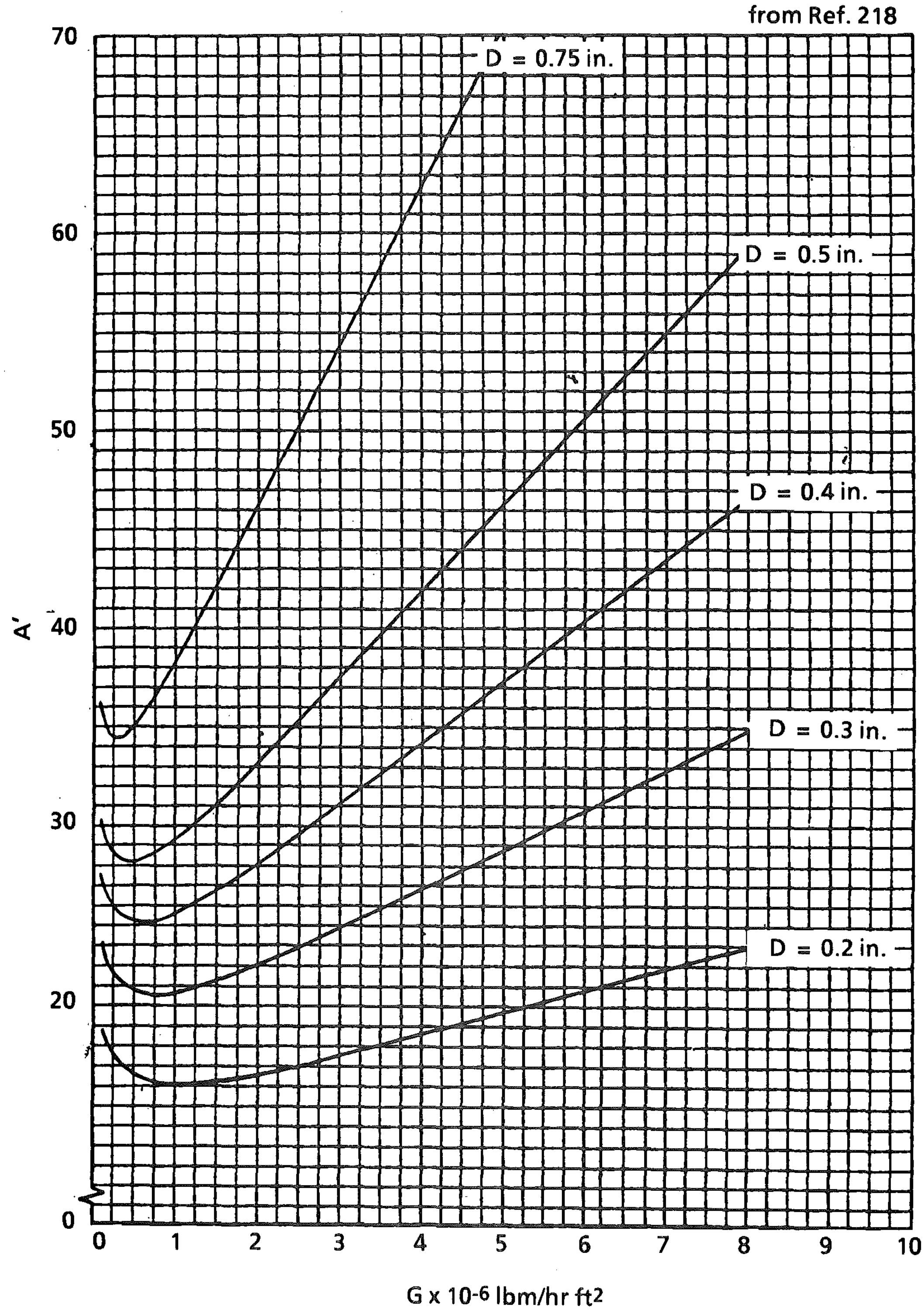
b. C′ for p = 560 psia
Figure 42. Continued.



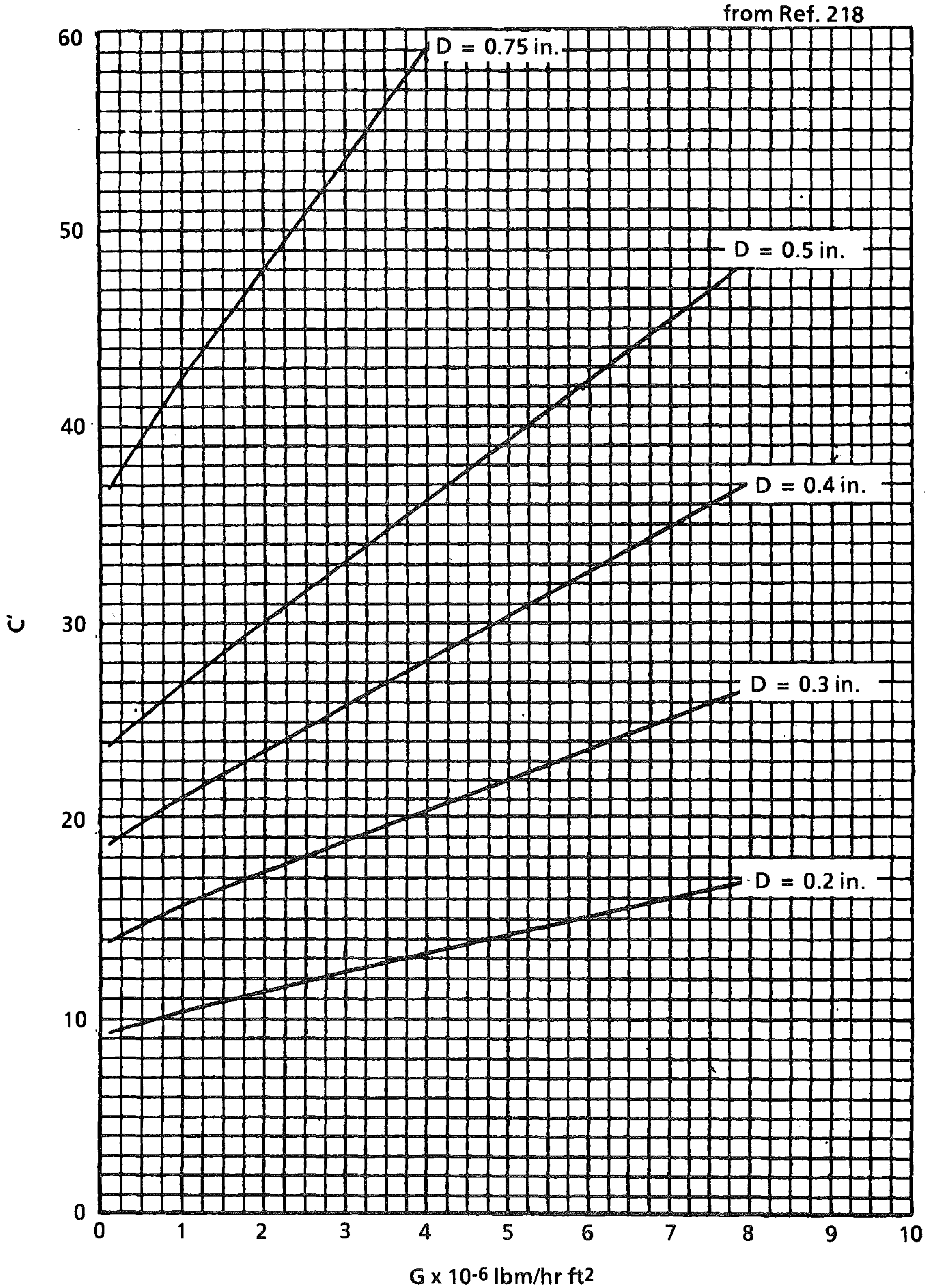
c. A′ for p = 1,000 psia
Figure 42. Continued.



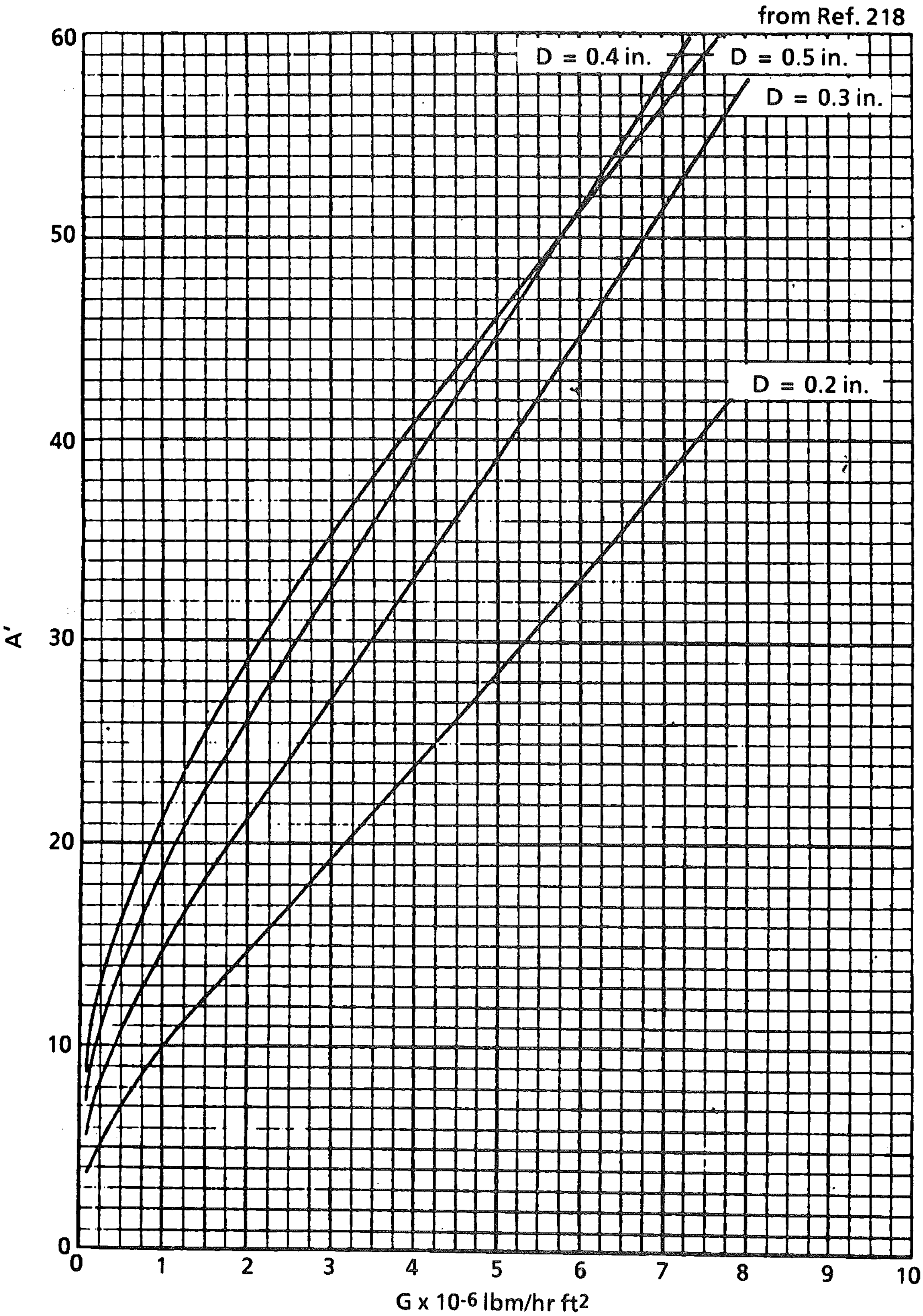
d. C′ for p = 1,000 psia
Figure 42. Continued.



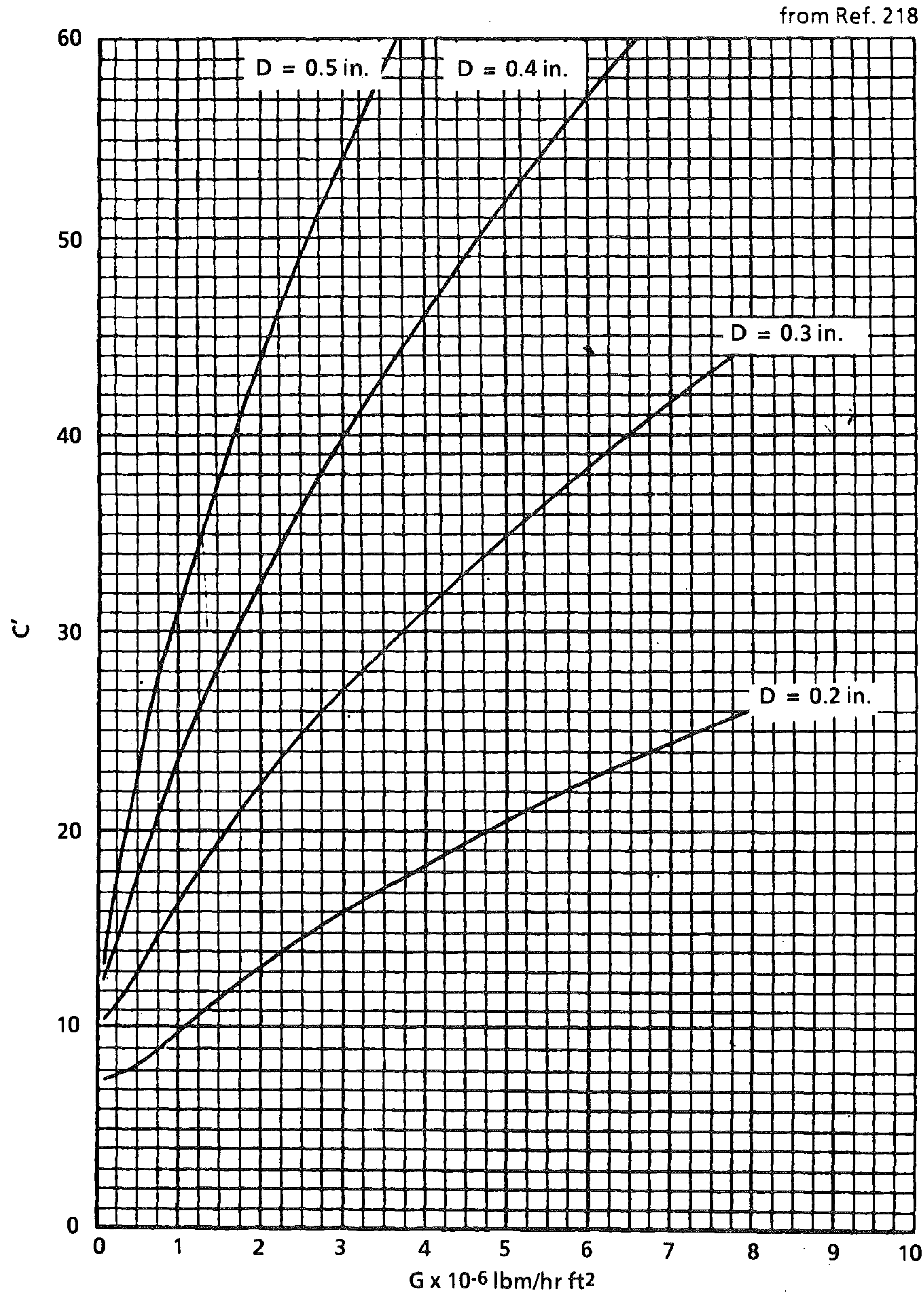
e. A′ for p = 1,550 psia
Figure 42. Continued.



f. C′ for p = 1,550 psia
Figure 42. Continued.



g. A′ for p = 2,000 psia
Figure 42. Continued.



h. C′ for p = 2,000 psia
Figure 42. Concluded.

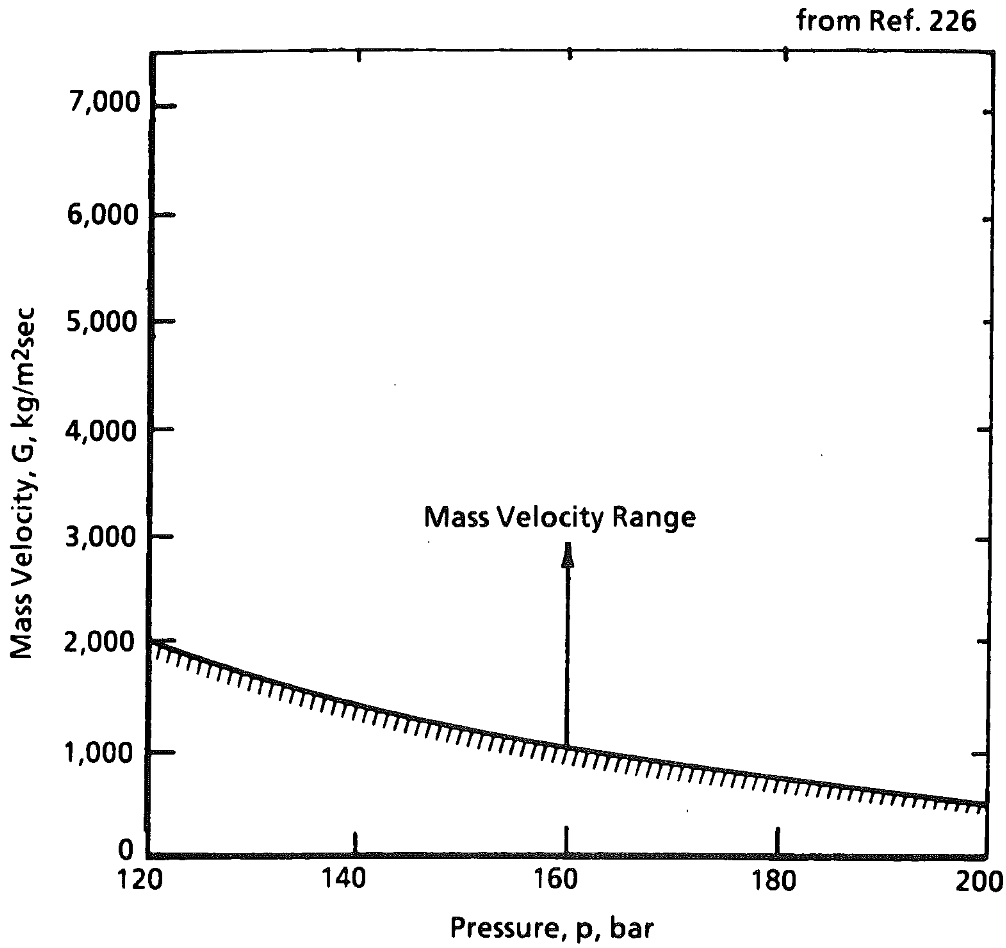


Figure 43. Mass velocity range for the Becker burnout correlation.

APPENDIX A

Table of Conversion Factors

Multiply	By	To Obtain	Parameter
ft^2	9.290E−02	m^2	A
ft/sec	3.048E−01	m/sec	V
ft	3.048E−01	m	D,L
Btu/lbm°F	4.187E+00	kJ/kg°C	c_p
Btu/ft^2hr	3.155E+00	W/m^2	q″,CHF,q/A
in.	2.540E+01	mm	d
lbm/ft^2hr	1.356E−03	kg/m^2sec	G
ft/hr^2	2.352E−08	m/sec^2	g,a
$Btu/ft^2hr°F$	5.679E+00	$W/m^2°C$	h
Btu/lbm	2.326E+00	kJ/kg	i, i_{fg}
Btu/ft hr°F	1.731E+00	W/m°C	k
psia	6.895E−02	bar	p
psia	6.895E+03	Pa	p
°F (differential)	5.556E−01	°C, K	ΔT_{sub}, ΔT_{sat}
ft^2/hr	2.581E−05	m^2/sec	α
lbm/ft hr	4.134E−04	kg/m sec	μ
lbm/ft^3	1.602E+01	kg/m^3	ϱ
lbf/ft	1.459E+01	N/m	σ
lbf/ft^2	4.788E+01	N/m^2	τ

°C = 5/9 (°F − 32)

K = °C + 273.15

APPENDIX B

Forced-Convection Boiling Experiments

Investigator	Ref	Yr	Organization	TYPE: Subcool (surf)	TYPE: Sat (bulk)	SURFACE: Tube	SURFACE: Annulus	SURFACE: Rod / Wire	SURFACE: Tube Bundle	SURFACE: Plate, Channel	SURFACE: Ribbon	Matl	COOLANT: Type
Monrad & Pelton	254	42	Carnegie Inst	x			x					copper	water
Davidson, et al.	255	43	Con. Edison	x	x				x			steel	water
Addoms	15	48	MIT	x	x			x				Pt,Ni,Al,SS,chromel	water
McAdams, et al.	24,256	48	MIT	x			x					SS	water
Knowles	147	48	NRC, Canada	x			x					SS	water
Dew	34	48	MIT	x			x					SS	water
Kreith & Summerfield	93	49	JPL/Cal Tech	x		x						SS	water
Kreith & Summerfield	22	49	JPL/Cal Tech	x		x						SS	var,water
Kaufman & Isely	257	50	NASA	x		x						Inconel	water
Buchberg, et al.	100	51	UCLA	x		x						SS	water
Gunther	27	51	JPL/Cal Tech	x							x	metal	water
McGill & Sibbitt	211	51	ANL/Purdue U	x		x						SS	water
Rohsenow & Clark	258	51	MIT	x		x						nickel	water
Jens & Lottes	259	52	Argonne N L	x	x	x				x		SS	water
Schweppe & Foust	82	53	UMI/Lehigh	x	x	x						copper	water
Ellion	138	54	JPL/Cal Tech	x			x					SS	water
Piret & Isbin	260	54	U of MN		x	x						copper	var,water
Clark & Rohsenow	23	54	MIT	x	x	x						L-nickel	water
Sabersky & Mulligan	261	55	Cal Tech	x		x						SS	water
McEwen, et al.	262	57	GE/AEC		x		x					Cu-Ni alloy	water
Powell	71	57	JPL/Cal Tech	x	x	x						SS	liq oxygen
DeBortoli & Masnovi	102	57	Westinghouse	x		x						nickel	water
Reynolds	263	57	MIT	x	x	x						nickel	water
Zenkevich, et al.	264	58	USSR	x	x				x			copper	water
Stein & Begell	117	58	Columbia U	x			x					aluminum	water
Lowdermilk, et al.	101	58	NASA	x	x	x						SS	water
Griffith, et al.	265	58	MIT, U of MI	x							x	?	water
Core & Sato	266	58	Aerojet	x		x	x					SS	var,water
DeBortoli, et al.	220	58	Westinghouse	x	x	x				x		Zircaloy-2,Ni,SS	water
Jacket, Roarty & Zerbe	79	58	Westinghouse	x	x	x				x		nickel, Zircaloy-2	water
Gambill & Greene	267	58	Oak Ridge NL	x	x	x						Cu,Inconel,Ni,Mo	water
Costello	236	59	U of WA	x			x					SS	water
Foltz & Murray	268	59	Goodyear		x	x						copper	Freon
Green, et al.	31	59	Westinghouse	x	x	x				x		SS,Ni,alloys	water
Zenkevich & Subbotin	269	59	USSR	x		x						SS	water
Schrock & Grossman	270	59	U of CA		x	x						SS	water
Zenkevich, et al	271	60	USSR	x					x			SS	water
Owens & Schrock	272	60	U of CA	x					x			SS	water
Gambill, et al.	115	61	Oak Ridge NL	x		x						Ni,Al,Cu	water
Aladyev, et al.	45	61	USSR	x	x	x						SS	water
Mendler, et al.	273	61	Westinghouse	x	x					x		SS	water
Bankoff & Mason	274	62	Northwestern	x						x		brass	water
Swenson, et al.	275	62	Babcock & Wil		x	x						SS	water
Swenson, et al.	120	62	Babcock & Wil		x	x						SS	water
Styrikovich, et al.	118	62	USSR	x	x	x						SS	water
Schaefer & Jack	276	62	NASA	x		x						Inconel	water
Papell	277	63	NASA	x		x						Inconel X	water
Judd & Wade	278	63	McMaster U	?	?		x					SS	water
Becker	279	63	Sweden		x				x			SS	water
Lee & Obertelli	83,280	63	UK AEEW	x		x						SS	water
Matzner	84	63	Columbia U	x		x						?	water
Styrikovich, et al.	281	63	USSR	x	x	x						SS	water
Janssen, et al.	215	63	GE	x	x		x					?	water
Bergles	85	63	MIT	x		x						SS	water
Tippets	28	64	GE	x	x					x		SS	water
Jiji & Clark	242	64	NYU, U of MI	x						x		SS	water
Becker & Hernborg	89	64	Sweden	x			x					SS	water
Alekseev, et al.	92	64	USSR	x	x	x	x	x				?	water
Bergles	143	64	MIT	x		x						SS	water
Bergles & Rohsenow	161	64	MIT	x	x	x	x					SS	water
Jeglic & Yang	245	65	NASA	x	x	x						Inconel X	water
Hosler	282	65	Westinghouse	x	x					x		SS	water
Durant, et al.	47	65	du Pont/SRL	x			x					SS	water
Feldmanis	283	65	AFFDL		x	x						copper	water
Stenning & Veziroglu	244	65	U of Miami	x		x						nichrome	Freon 11
Ornatskii & Vinyarskii	232	65	USSR	x		x						Cu alloy,brass/Ag	water
Waters, et al.	284	65	GE	x	x	x						Inconel	water
Rousar & Van Huff	285	66	Aerojet/JPL	x		x						Inconel 718,SS	H2O2,water
Owhadi	286	66	OK State U	x	x	x						Inconel 600	water
Todreas & Rohsenow	287	66	AEC/MIT	x	x	x						aluminum	water
Moeck, et al.	288	66	Can/Colum. U	x	x		x					?	water
Maulbetsch & Griffith	289	66	MIT	x		x						SS	water
Larsen, et al.	290	66	Westinghouse	x	x		x,square					Hastelloy-C®	water
Bergles & Suo	291	66	MIT	x	x	x						SS	water
Brown	40	67	MIT				x					SS	water
Frost & Kippenhan	54	67	UT/UWA	x			x					SS	water
Glickstein & Whitesides	292	67	Pratt&Whitney	x	x	x						Inconel	var
Weisman, et al.	293	68	Westinghs/PNL	x					x			Inconel	water
Fiori & Bergles	20	68	MIT	x		x	x					SS	water
Markels & Durfee	142	68	ARC, VA	x			x					SS	water
Lopina & Bergles	294	69	MIT,USAF Ac.	x		x						nickel	water
Borishanskiy & Fokin	64	69	USSR	x	x	x	x	x				SS,nichrome	var,water
Coffield, et al.	209	69	U Pittsburgh	x		x						SS	Freon-113
Roemer	127	69	Stanford U	x				x				SS	water
Miropol'skiy & Pikus	295	69	USSR	x	x	x						SS	water
Tolubinskiy, et al.	296	69	USSR	x	x		x					SS	water
Bergles & Dormer	173	69	MIT	x		x						SS,nickel	water
Peskov, et al.	297	69	USSR	x	x	x						SS	water
Zenkevich, et al.	90	70	USSR	x	x	x	x					SS	water

Investigator	Treatment [1]: dis	dei	dem	deg	gas	untr	nwtd	Flow Orientation	HEATING: Coal,Gas Fired	Elect Htr	Elect Resis	Liquid Metal	Steam, Fluid
Monrad & Pelton							x	vert					x
Davidson, et al.						x		vert	x				
Addoms							x	horiz			dc		
McAdams, et al.	x			x	x			vert			dc		
Knowles						x		vert			x		
Dew	x			x				vert			dc		
Kreith & Summerfield							x	horiz,vert			x?		
Kreith & Summerfield							x	vert			x?		
Kaufman & Isely							x	horiz,vert			x?		
Buchberg, et al.	x			x	x			vert			dc		
Gunther	x			x				vert? [9]			dc		
McGill & Sibbitt				x				vert			x		
Rohsenow & Clark	x	x		x				vert			dc		
Jens & Lottes						x		vert,inclin			ac		
Schweppe & Foust							x	vert		x			
Ellion	x			x				vert			ac,dc		
Piret & Isbin	x							vert		x			
Clark & Rohsenow	x	x		x				vert			dc		
Sabersky & Mulligan	x							not defined			dc		
McEwen, et al.		x		x				horiz			dc		
Powell								vert?			x?		
DeBortoli & Masnovi				x	x			vert			x?		
Reynolds	x	x		x				vert			dc		
Zenkevich, et al.	x							vert			x		
Stein & Begell		x						vert			dc		
Lowdermilk, et al.	x			x	x			vert			ac		
Griffith, et al.							x	vert			x		
Core & Sato	x							vert			dc		
DeBortoli, et al.		x	x	x				vert,inclin			dc		
Jacket, Roarty & Zerbe		x	x	x				vert,inclin			dc		
Gambill & Greene						x		horiz			ac		
Costello	x							vert			ac		
Foltz & Murray								vert					x
Green, et al.							x	vert,inclin			dc		
Zenkevich & Subbotin	x							vert			x?		
Schrock & Grossman	x			x				vert			ac		
Zenkevich, et al	x							vert			x?		
Owens & Schrock				x				vert			ac		
Gambill, et al.							x	horiz			x?		
Aladyev, et al.							x	vert			ac,dc		
Mendler, et al.		x	x	x				vert			dc		
Bankoff & Mason	x			x				horiz					x
Swenson, et al.			x	x				vert			ac		
Swenson, et al.		x		x				vert			ac		
Styrikovich, et al.							x	vert			ac		
Schaefer & Jack	x							vert			dc		
Papell	x							horiz?			ac		
Judd & Wade						x		vert			x		
Becker							x	vert			dc		
Lee & Obertelli							x	vert			dc		
Matzner							x	vert			dc		
Styrikovich, et al.							x	vert			ac		
Janssen, et al.			x					vert			x?		
Bergles	x	x		x				horiz			dc		
Tippets							x	vert		x			
Jiji & Clark	x			x				vert			dc		
Becker & Hernborg							x	vert			dc		
Alekseev, et al.							x	various			x		
Bergles		x		x				vert			x		
Bergles & Rohsenow	x	x		x				horiz			dc		
Jeglic & Yang		x		x				vert			ac		
Hosler							x	horiz			dc		
Durant, et al.		x						vert			dc		
Feldmanis	x							horiz		x			
Stenning & Veziroglu								horiz?			dc		
Ornatskii & Vinyarskii							x	vert?			dc		
Waters, et al.							x	horiz,vert			dc		
Rousar & Van Huff		x						vert			dc		
Owhadi	x			x				vert			dc		
Todreas & Rohsenow							x	not defined			dc		
Moeck, et al.							x	vert			dc		
Maulbetsch & Griffith		x		x				horiz?			dc		
Larsen, et al.							x	vert			dc		
Bergles & Suo	x		x	x				vert			dc		
Brown	x			x				horiz			dc		
Frost & Kippenhan	x?							vert			dc		
Glickstein & Whitesides								vert			dc		
Weisman, et al.		x		x				vert			x?		
Fiori & Bergles	x	x		x				vert			dc		
Markels & Durfee		x						vert			dc		
Lopina & Bergles	x	x		x				horiz?			dc		x
Borishanskiy & Fokin	x							horiz,vert			x?		
Coffield, et al.								vert			dc		
Roemer			x	x				vert			dc		
Miropol'skiy & Pikus							x	not defined			dc		
Tolubinskiy, et al.	x			x				vert			x?		
Bergles & Dormer	x		x	x				horiz			dc		
Peskov, et al.	x							vert			ac		
Zenkevich, et al.							x	not defined			ac		

Investigator	EFFECTS STUDIED: dTsub [2]	Press	Vel [3]	Burnout crit pt	Gas Contnt	Accel	L/D [4]	Wall Matl	Dh [5]	Wall Thk	Rough-ness	Press Drop	Uni/Nonun Ht Flux	Stability	Transient	Optical	Other
Monrad & Pelton			x						x								
Davidson, et al.	x	x	x				x			x		x					
Addoms	x	x	x	x			x	x								x	
McAdams, et al.	x	x	x	x	x											x	
Knowles	x		x	x			x		x							x	
Dew	x	x	x				x		x			x				x	
Kreith & Summerfield	x	x	x								x	x					
Kreith & Summerfield	x	x	x									x					
Kaufman & Isely		x	x														
Buchberg, et al.	x	x	x	x	x							x					x
Gunther	x	x	x	x												x	
McGill & Sibbitt	x	x	x	x			x					x					
Rohsenow & Clark	x	x	x									x					
Jens & Lottes	x		x	x			x					x					
Schweppe & Foust	x		x				x		x			x					x
Ellion	x	x	x	x	x											x	
Piret & Isbin			x														
Clark & Rohsenow	x	x	x	x													
Sabersky & Mulligan		x										x					
McEwen, et al.			x	x													
Powell	x	x	x	x			x										
DeBortoli & Masnovi			x	x	x							x					
Reynolds	x	x	x	x													
Zenkevich, et al.	x	x	x	x													
Stein & Begell	x		x				x		x				x				
Lowdermilk, et al.	x	x	x	x			x		x			x					
Griffith, et al.	x	x	x				x									x	
Core & Sato	x	x	x	x													
DeBortoli, et al.	x	x	x	x	x		x	x	x				x				
Jacket, Roarty & Zerbe	x		x	x			x	x	x		x						
Gambill & Greene	x	x	x	x		x	x	x			x	x					x
Costello	x		x									x				x	x
Foltz & Murray		x	x				x		x			x					
Green, et al.	x	x	x	x	x		x	x	x								
Zenkevich & Subbotin	x	x	x	x	x												
Schrock & Grossman		x	x				x					x					
Zenkevich, et al	x	x	x	x													
Owens & Schrock	x	x	x				x		x			x					
Gambill, et al.	x	x	x	x			x		x			x					x
Aladyev, et al.	x	x	x	x			x			x	x		x				
Mendler, et al.	x	x	x	x			x		x			x					
Bankoff & Mason	x		x													x	x
Swenson, et al.			x	x													x
Swenson, et al.			x	x									x				
Styrikovich, et al.	x	x	x	x									x				
Schaefer & Jack	x	x	x	x			x			x		x	x				
Papell	x	x	x														
Judd & Wade									x			x					x
Becker		x	x	x													x
Lee & Obertelli	x	x	x	x			x		x	x			x				
Matzner	x		x	x			x		x	x							
Styrikovich, et al.	x		x	x			x						x				
Janssen, et al.	x	x	x	x			x		x		x		x				x
Bergles	x	x	x	x			x		x					x			
Tippets	x		x	x									x			x	
Jiji & Clark	x	x	x													x	x
Becker & Hernborg	x	x	x	x								x	x				
Alekseev, et al.	x	x	x	x			x										x
Bergles			x	x										x			
Bergles & Rohsenow	x		x				x										
Jeglic & Yang	x	x	x				x							x		x	
Hosler	x	x	x													x	
Durant, et al.	x	x	x	x					x		x						
Feldmanis						x						x					
Stenning & Veziroglu			x									x		x			
Ornatskii & Vinyarskii	x	x	x	x			x		x				x				
Waters, et al.	x	x	x	x								x					
Rousar & Van Huff	x	x	x	x			x	x	x	x		x					x
Owhadi			x			x						x					x
Todreas & Rohsenow	x	x	x	x			x						x				
Moeck, et al.	x		x	x					x								x
Maulbetsch & Griffith	x	x	x	x			x		x			x		x			
Larsen, et al.	x	x	x	x													
Bergles & Suo		x	x		x		x									x	
Brown	x	x			x						x					x	
Frost & Kippenhan	x		x	x												x	x
Glickstein & Whitesides		x	x	x													
Weisman, et al.	x	x		x									x				
Fiori & Bergles	x	x	x	x			x		x	x						x	
Markels & Durfee	x	x	x	x			x		x								x
Lopina & Bergles	x	x	x				x					x					x
Borishanskiy & Fokin	x	x	x	x						x					x		
Coffield, et al.	x	x	x	x													
Roemer				x											x	x	x
Miropol'skiy & Pikus	x	x	x	x													x
Tolubinskiy, et al.	x	x	x	x			x		x								
Bergles & Dormer	x	x	x	x			x	x	x			x					
Peskov, et al.	x	x	x	x			x										
Zenkevich, et al.	x	x	x	x			x		x				x				

Investigator	PARAMETER RANGE: dTsub deg-C [2]		L/D [4]	Press bar	qdot [6] MW/sq-m	Vel m/s	mdot [7] kg/s	G [8] Mg/m2 s	Comments
Monrad & Pelton	?		39,90	?	0.02-0.99		.18-1.1		counterflow
Davidson, et al.	0-117	E	211-722	34.5-228	23-120	.6-21	.04-.65	.1-1.5	
Addoms	0-82	L	46	1-170	0-7.9	0-3			
McAdams, et al.	11-83	I	15	2.1-6.2	0-6.3	.3-3.7			
Knowles	0-100	I	7.6-76	2.8	0-7.7			.77-3	
Dew	11-83	A	8.3,23.5	2.1-6.2	.03-6.4	.3-11.3			
Kreith & Summerfield	50-167	L	30-73	1.1-14	1.2-4.6	1.8-4	.15-.67		
Kreith & Summerfield	?		30	2.1-28	.5-4.9	6.4-12.5			follow-on to previous study
Kaufman & Isely	41-68	I	50	.33-6.8	.2-2		.024-.1		
Buchberg, et al.	0-131	L	111	6.9-173	0-11.9			1.3-10.4	density study
Gunther	12-142	E	32,48	1-11.3	1.3-35.9	1.5-12.2			
McGill & Sibbitt	3-353	I	21-168	18.3-208	.1-14	.18-12.2			
Rohsenow & Clark	28-139	L	52	104, 138	0-11			up to 7.6	
Jens & Lottes	0-83	I	8.5-257	1	0-.9		.002-.05		
Schweppe & Foust	?		8-19.5	?	.12-.65			.49-6.5	turbulence studied
Ellion	28, 56	?	1.3-8.3	1.1, 4.1	.07-3.2	.34, 1.5			
Piret & Isbin	0-2.2	I	54	1	0-.17	.23-1.07			
Clark & Rohsenow	0-167	I	52	6.9-138	0-9.4			.01-.36	
Sabersky & Mulligan	44	I	6.7	4.5, 18.3	1.6-3.3		.33		friction evaluated
McEwen, et al.	55.6	I	656	104	.32-1.25			.8-1.9	
Powell	?		31-371	17.6-74	0-4.1?			2.1-27.3	
DeBortoli & Masnovi	74-78	I	64.5	138	0-4.3			.75-2.8	hydrogen concentration
Reynolds	0-179	I	87	34.5-104	0-9	1.5, 3		1.17-2.9	
Zenkevich, et al.	10-40	?	40	183-213	?	.64-4.6			
Stein & Begell	?		160-480	?	.12-3.5	1.2-13.7	.32-7.5		
Lowdermilk, et al.	0-78	I	25-250	1-6.9	0-41.5	.03-29.9			
Griffith, et al.	5-83	I	8	34.5-104	.08-8.5	6.1-9.1			void fraction measured
Core & Sato	0-213	L	42	1-34.5	0-8.4	.3-6.9			outer wall heated
DeBortoli, et al.	0-333	I	21-365	34.5-207	0-31			.03-10.6	various other data included
Jacket, Roarty & Zerbe	0-186	I	64,140	138	0-5.7			.21-4.5	
Gambill & Greene	0-88	E	2-20	6.5-64	0-172	3.7-29.3			straight & swirl flow
Costello	13-57	I	35	1.2	0-2.6	.46-1.2			vapor thickness measured
Foltz & Murray	0	I	21-495	5.2-7.2	.01-.08			.26-.69	
Green, et al.	0-308	I	33-530	41.4-138	.37-14.7			.04-10.5	
Zenkevich & Subbotin	10-100	?	28-165	142-213	0-13.5			.83-5	
Schrock & Grossman	0	I	120,160	3.5-27.6	.32-2.5			.98-3.4	
Zenkevich, et al	10-40	?	40	183-213	?	.64-4.6			
Owens & Schrock	0-141	I	88-127	3.6-27.9	.67-4			.83-5.3	
Gambill, et al.	0-144	E	45-70	2-15.2	2.5-116	4.5-47.5			swirl flow; twisted tapes
Aladyev, et al.	0-180	E	19.5	20.7-207	0-8.7			.4-40	
Mendler, et al.	11-56	I	68-149	55.2-138	.16-2			.48-.56	
Bankoff & Mason	18-73	L	-	1	?	.27-2.2			bubble growth; jet impingement
Swenson, et al.	0	L	176	207	.28-.57			.95, 1.36	straight & swirl flow
Swenson, et al.	0	L	164	138	.22-1.8			.68-1.76	
Styrikovich, et al.	0-103	L	27	101-183	0-7.8			.4-2	
Schaefer & Jack	105-191	A	28-61.5	4.8-21.2	0-133	22-62		22-62	
Papell	100-146	L	21	2.6-12.4	.6-2.6	1.2-3.8			
Judd & Wade	?		16-96	?	?	?			eccentricity effects
Becker	0	L	64	2.5-10.8	.5-1.2		.032-.33		
Lee & Obertelli	2.8-139	I	20-360	38.6-110	0-7.7			.4-4	
Matzner	9-210	I	26-151	69	0-8.1			.68-18.6	
Styrikovich, et al.	0-42	L	20-118	101	0-9.9			.4-2	
Janssen, et al.	0-33	E	33-600	41.4-100	0-5.5			.2-8.4	eccentricity
Bergles	0-58	E	5-35	1.4-5.9	0-44.7			1.5-24.3	
Tippets	8-144	I	17.6	68.7-72.8	0-3.6			.24-1.9	
Jiji & Clark	28-167	I	20	13.8-69	.44-4.7	.3-1.8			flow temp probe measurement
Becker & Hernborg	60-205	I	81	8.3-36.8	0-3			.07-.96	
Alekseev, et al.	?		40,67	29.3-207	0-11			0-5.6	multiple flow configurations
Bergles	103	I	24	1.5	0-12.6	2.1-8.5			
Bergles & Rohsenow	0-108	I	20-58	17.9	0-18.9	0-17.4			
Jeglic & Yang	0-108	I	100	.2-6.9	0-2.65	.7-2.5			
Hosler	0-28	I	102	10.4-41.4	0-3.15			.14-1.4	primarily photo data
Durant, et al.	38-88	?	48-96	2.1-4.1	7-11.3	3-8.5			knurled surfaces
Feldmanis	0	I	118	.03-.2	?	.03-22	.001-.0014		low gravity flow
Stenning & Veziroglu	32	I	254	5.2	?		.003-.018		
Ornatskii & Vinyarskii	50-190	?	7-35	11.2-61.9	0-343			.01-.09	highest heat flux ever achieved
Waters, et al.	0-278	I	327	70-105	2-5.4			6.45-9.6	upstream burnout
Rousar & Van Huff	151-331	A	23,31	69-207	0-45.7	11.6-45.7			
Owhadi	0-51	I	245	1	.06-.25		.009-.039		coiled flow
Todreas & Rohsenow	0-189	I	140-224	2.6-15.9	?			.68-2.7	
Moeck, et al.	11-119	I	356-579	69	0-2.7			.68-6.8	stripping rings,eccentricity
Maulbetsch & Griffith	108-139	I	25-200	2.1-5.8	0-22			3-6.8	
Larsen, et al.	7-78	I	118	105-160	0-4.9			.66-4.1	
Bergles & Suo	0-139	I	60-240	34.5-69	?			.54-5.4	
Brown	6-100	?	20	2.1-6.2	0-1.6	0-1.7			bubble detector
Frost & Kippenhan	44-83	I	23	2.1	0-6.3	.24-1.2			varied surface tension
Glickstein & Whitesides	0-211	L	98.5	10.4-55.2	.27-2.1			.2-4.5	nucleate & film boiling
Weisman, et al.	22-128	I	28-59	125-160	0-2			.68-4	
Fiori & Bergles	28-61	E	15-54	1.7-6.2	3.15-17.3			.68-10.2	
Markels & Durfee	0-258	I	31-88	1.4-104	0-5.4			.2-2.7	applied a-c field
Lopina & Bergles	?		69-76	2.1-6.9	?		.013-.18		nonboiling, swirl flow
Borishanskiy & Fokin	0-58	?	8.4-33	1-24.6	0-4.9	0-2.5			unsteady heat application
Coffield, et al.	2.8-24	I	74.6	9.1-21.9	0-.57			1.3-5.6	DNB detector
Roemer	1.8-2.4,40	?	5.3-16	?	0-2.3	1			power transients, aging studies
Miropol'skiy & Pikus	0-106	?	?	98-215	0-4			.1-2	curved channels
Tolubinskiy, et al.	0-143	I	7.7-750	80-220	0-7			.05-.8	outer wall heated
Bergles & Dormer	58-141	I	24-195	2.1-5.5	0-17.3	1.5-18.3			
Peskov, et al.	161-308	I	31-375	101-203	.58-4.9			.82-5	
Zenkevich, et al.	0-60	I	93-730	39.3-196	0-4.5			.3-2.8	

APPENDIX B

Continued

Investigator	Ref	Yr	Organization	TYPE Subcool (surf)	TYPE Sat (bulk)	SURFACE Tube	Annulus	Rod / Wire	Tube Bundle	Plate, Channel	Ribbon	Matl	COOLANT Type	Treatment [1] dis	dei	dem	deg	gas	untr	nwtd	Flow Orientation
Tolubinskiy & Kostanchuk	38	70	USSR	x						x		SS	water	x			x				vert
Tolubinskiy, et al.	139	70	USSR	x	x		x					SS	water							x	vert
Sterman & Nekrasow	298	70	USSR		x	x						steel	water							x	vert
Stefanovic, et al.	243	70	Yugoslavia		x	x	x					?	water	x							vert
Osmachkin & Borisov	299	70	USSR	x	x	x						SS	water							x	vert
Gaddis & Hall	300	70	U of Manc,Eng	x						x		copper, Ni plated	water				x				horiz?
Johnson	301	70	U of CA,Berkl	x							x	nickel/iron	water							x	horiz
Staub & Walmet	175	70	GE	x		x				x		?	R-22,water							x	vert
Kawamura, et al.	302	70	Japan	x	x		x				x	SS, nickel	water							x	vert
Ornatskii, et al.	208	71	USSR	x	x	x						SS	water			x	x				not defined
Veziroglu & Lee	137	71	U of Miami	x?		x						nichrome	Freon-11								vert
Motley, et al.	67	72	Westingh/CMU	x					x	x		?	Freon,water							x	not defined
Mattson	303	72	U of MI	x						x		SS?	Freon-113								horiz
Becker, et al.	226	72	Royal I, Swe	x		x						SS	water							x	vert
Ahmad & Groeneveld	66	72	AECL, Can	x			x					nichrome V	Freon-12								not defined
Abdelmessih, et al.	26	72	U of Toronto	x							x	SS	water	x							vert
Lopina & Bergles	113	73	MIT	x		x						nickel	water			x	x				not defined
Plummer, et al.	304	73	GE		x	x						Inconel X-750	water							x	vert?
Purcupile, et al.	68	73	Westingh/CMU	x		x						?	R-11								vert
Knoebel, et al.	227	73	Col U/du Pont	x			x			x		SS,aluminum	water,D20	x	x		x				vert
Tolubinskiy & Matorin	63	73	USSR	x		x						?	var mixes								not defined
Megerlin, et al.	116	74	WGr,Rayth,ISU	x		x						SS	water	x		x	x				not defined
Katto & Monde	305	74	Japan		x						x	Cr-SS	water							x	horiz
Hughes	21	74	Penn State	x							x	nichrome	Freon 113								horiz
Levitan & Lantsman	223	75	USSR	x	x	x						?	water							x	not defined
Ornatskiy, et al.	306	75	USSR	x			x					?	water							x	not defined
Ornatskiy, et al.	114	75	USSR	x	x		x					?	water							x	vert
Iloeje, et al.	307	75	MIT		x	x						Inconel X-750	water							x	vert
Kovalev, et al.	49	76	USSR	x				x				copper	water							x	vert?
Merilo	94	77	Chalk R.,Can	x	x	x						Inconel 600	Freon12,water							x	horiz,vert
Spencer & Rousar	308	77	NASA	x		x						Monel K-500,Inconel	liq. oxygen								vert
Ornatskiy & Sharayevskiy	248	77	USSR	x			x					SS	water				x				not defined
Cumo, et al.	309	77	Italy	x							x	SS	Freon								vert
Lung, et al.	310	77	Germany	x			x					Inconel 600	water			x					vert
Eddington & Kenning	311	78	Oxford U	x						x		SS	water				x				horiz
Eddington, et al.	312	78	Oxford U/USSR	x						x		brass	water	x	x		x				horiz
Cheng, et al.	124	78	U of Ottawa	x	x	x						copper	water	x							vert
Kutateladze & Malenkov	313	78	USSR		x		x					microporous matl	var,water	x							vert?
Monde & Katto	314	78	Japan	x	x					x		copper	Freon,water	x?							vert
Yucel & Kakac	140	78	Turkey	x						x		SS	water		x						horiz,vert
Drizius, et al.	315	78	USSR	x		x						copper	water	x							not defined
Doroschuk, et al.	316	78	USSR	x	x	x	x					?	var							x	vert
Ragheb, et al .	317	78	U of Ottawa	x	x					x		copper	water							x	not defined
Mel'nikov, et al.	107	79	USSR	x	x		x					SS	water							x	vert
Chen, et al.	318	79	U of Ottawa	x		x						SS,Inconel	water			x					not defined
Romberg & Harris	246	79	Australia	x?			x					SS	Freon 113								not defined
Ragheb & Cheng	319	79	U of Ottawa	x	x					x		copper	water							x	vert?
Bennett & Chen	320	80	Lehigh U		x	x						Inconel	glycol,water	x							vert
Mori, et al.	321	80	Japan	x				x		x		nickel,glass	water/oil							x	horiz
Petukhov, et al.	322	80	USSR		x	x						SS	liq helium								vert
Koba, et al.	323	80	USSR	x	x	x						steel	water	x							horiz?
Tolubinskiy, et al.	250	80	USSR	x			x					?	water							x	not defined
Yilmaz & Westwater	324	80	HTRI/U of IL		x	x						copper	Freon-113								vert
Ornatskiy, et al.	119	80	USSR	x			x					?	water							x	vert
Jain, et al.	156	80	U of IL	x			x					SS	R-113								vert
Ornatskiy & Sharayevskiy	249	80	USSR	x			x					SS	water							x	not defined
Fisenko, et al.	99	80	USSR	x		x						SS	water				x	x			vert
Leontiev, et al.	121	81	USSR	x	x	x						SS	water			x					horiz
Zeigarnik, et al.	96	81	USSR	x						x		copper	water			x					horiz
Yao	325	81	Carnegie-Mell	x	x		x					SS	var,water	x		x					vert
Styrikovich, et al.	326	81	USSR	x	x	x						SS	water			x?					horiz
Ragheb, et al.	123	81	U of Ottawa	x	x	x						Cu,Inc,Al,Zircaloy	water							x	vert?
Ramamohan, et al.	327	81	India	x			x					platinum	gly,alc/water							x	vert?
Jensen & Bergles	95	81	U of WI,IA St	x	x	x						SS	R-113								vert
France, et al.	328	81	Argonne N L	x		x						chromoly steel	water							x	vert
Agrawal, et al.	329	82	India		x	x						SS	R-12								horiz
Aounallah, et al.	330	82	Oxford U		x	x						cupro-nickel	water							x	vert
Del Valle & Kenning	81	82	Oxford U	x						x		SS	water	x			x				horiz
Fukuyama & Hirata	331	82	Japan	x		x						SS	Freon 113								horiz
Katto & Ashida	332	82	Japan	x	x	x						SS	R-12								vert
Leung, et al.	44	82	AECL,Can/UCA	x			x					Inconel 600	water						x		vert
Mishima & Ishii	333	82	Japan/ANL	x	x		x					SS	water		x						vert
Nishikawa, et al.	190	82	Japan	x	x	x						SS	Freon 22								vert
Papaioannou & Koumoutsos	61	82	Greece		?			x				bronze	water/polymer							x	horiz
Rogers, et al.	240	82	U of Ottawa	x	x		x					Inconel 718	water							x	vert
Katto & Yokoya	32	82	Japan	x	x	x						SS	R-12								vert
Sekoguchi, et al.	334	82	Japan	x	x	x						?	water	x			x				vert
Viannay & Karian	335	82	France	x			x			x		steel,copper	var,water							x	not defined
Wei & Maa	62	82	Taiwan				x					SS,copper	water/polymer	x							vert
Ueda, et al.	336	83	Japan	x	x	x						copper	Freon-113								vert
Sharon, et al.	337	83	Northwestern	x	x		x					SS	water		x						vert
Bilicki	55	83	Poland		x	x						SS	Freon 21								horiz,vert
Weede & Dhir	145	83	UCLA	x		x						Inconel 600	Freon 113								vert
Klimenko & Sudarchikov	338	83	USSR	x	x	x						SS	liq nitrogen								vert
Kataoka, et al.	125	83	Japan	x	x			x				platinum	water	x	x						vert
Kalayda, et al.	103	83	USSR	x			x					SS	water				x	x			vert?
Baines, et al .	339	84	MIT	x	x					x		copper	water,Freon							x	inclined
Katto & Ohno	189	84	Japan	x		x						SS	R-12								vert
Kozhelupenko, et al.	340	85	USSR	x			x					SS	water	x							horiz?
Hino & Ueda	341	85	Japan	x			x					SS	Freon 113								vert

Investigator	Coal,Gas Fired	HEATING Elect Htr	Elect Resis	Liquid Metal	Steam, Fluid	EFFECTS STUDIED dTsub [2]	Press	Vel [3]	Burnout crit pt	Gas Contnt	Accel	L/D [4]	Wall Matl	Dh [5]	Wall Thk	Rough-ness	Press Drop	Uni/Nonun Ht Flux	Stability	Transient	Optical	Other	PARAMETER RANGE dTsub deg-C [2]	L/D [4]	Press bar	qdot [6] MW/sq-m	Vel m/s	mdot [7] kg/s	G [8] Mg/m2 s	Comments
Tolubinskiy & Kostanchuk			dc			x	x	x													x	x	5-60 ?	?	1-10	.05-1	.08-.2			bubble growth measurements
Tolubinskiy, et al.			x?			x	x	x	x			x		x									0-313 ?	67-1500?	50-200	0-6.5			.25-3	varied heating modes
Sterman & Nekrasow			ac				x	x	x			x											0 ?	?	29.4-98	.5-10.5			.83-3.4	
Stefanovic, et al.			dc					x													x	x	0? ?	27,69	1	.07-.27		.26-.34		b.l. probe measurements
Osmachkin & Borisov			dc														x					x	?	519	70	.71		6.9		
Gaddis & Hall		x				x		x								x					x		20-60 L	63	?	0-.15	.1-.5			drilled cavity
Johnson			x			x	x	x	x			x								x	x		5.6-62 ?	7.4	1-138	0-.05	.3, 4.3			
Staub & Walmet			?			x	x	x						x			x						4-91 L	25.8,150	.1-69	0-.08?			.18-2.8	void fraction measurements
Kawamura, et al.			x?			x		x	x						x					x	x		0-70 ?	2.2	1	0-15	0-2			transient power application
Ornatskii, et al.			x?			x	x	x	x			x		x									0-87 ?	30-150	9.8-167	0-17.8			1.5-8	
Veziroglu & Lee			dc					x									x		x				?	188,254	5.5	?		.0009-.018		parallel channels
Motley, et al.			dc				x	x	x					x				x					?	40,227	11.2-166	0-4.5			2-5.4	
Mattson			x?			x	x	x	x												x		22-66 E	?	7.2-24.3	.09-.96			1.6-5.6	
Becker, et al.			dc			x	x	x	x			x											8-272 I	200-500	100-200	.14-3.5			.16-7.6	
Ahmad & Groeneveld		x				x	x	x															.4-10 I	155,222	6.2-16.6	0-3.6			.14-3.3	
Abdelmessih, et al.			x					x													x		1.9 ?	2.7?	?	.19-.46	.9-2.3			
Lopina & Bergles			dc				x	x									x				x	x	?	D=4.93mm	2.1-3.5	0-11	2.9-7.6			swirl flow
Plummer, et al.			x?			x	x	x	x													x	0 I	445	6.9-69	0-2.8			.07-.34	fouling evaluated, film boiling
Purcupile, et al.			?			x	x	x	x														18-28 I	90	10.8-25	0-1.2			1.4-8.8	
Knoebel, et al.			dc			x	x	x	x			x	x	x				x					3.9-89 L	40-64	2.1-6.6	0-20.5	4.6-18.3			bowing of heater important
Tolubinskiy & Matorin			?			x	x	x	x			x		x									?	10-15	3.3-13.2	0-13.5	2.5-10			binary mixes
Megerlin, et al.			ac			x	x	x	x								x					x	82-203 I	9.5	2-17.4	0-31.5			2.1-4	tube inserts
Katto & Monde			ac					x	x												x		3 L	-	1	0-18	0-60			jet impingement
Hughes			dc			x		x	x		x				x	x							28-61 I	11	7.6-10.4	0-1.95	.3-1.3			straight and curved flow
Levitan & Lantsman			?			x	x	x	x			x											?	188-375	49-186	0-2.75			1-3	
Ornatskiy, et al.			x			x	x	x	x			x						x					38-337 I	75-162.5	98.1-187	0-5.8			.5-1.7	
Ornatskiy, et al.			dc			x	x	x	x			x										x	0-159 I	53-150	98.1-187	0-7.25			.5-1.7	with & without swirl
Iloeje, et al.			x?					x	x											x		x	0 I	8.1	69	0-2.75			.07-.34	
Kovalev, et al.	*note					x		x	x							x						x	15-70 I	-	1	0-10.4	.02-.14			finned geoms; electron bomb htg
Merilo			dc			x	x	x	x			x											0-168 I	97-387	10.5-96.6	0-2.3			.7-5.7	
Spencer & Rousar			dc			x	x	x	x			x											?	3.6-204	173-345	2-89.7			24-122	
Ornatskiy & Sharayevskiy			dc			x	x	x	x													x	10-120 I	13.3	1-4	0-3.5	1-2			acoustic emissions measured
Cumo, et al.			dc			x	x	x													x		5-55 L	20	10.7-21.8	.06-.28			.14-.37	vapor thickness measured
Lung, et al.			dc			x	x	x															10-40 A	50	0.5-7	0-3	0.5-8			
Eddington & Kenning			dc														x				x		3-6 I	?	1.2	0-.21	.15-.3			Del Valle/Kenning rig
Eddington, et al.			dc														x				x		3 I	?	1.2	0-.11	.15			Del Valle/Kenning rig
Cheng, et al.		x				x		x	x														0-28 I	8.5	1	0-2			.068-.21	
Kutateladze & Malenkov								x		x									x				0 L	?	?	0	?			bubble stability
Monde & Katto		x				x		x	x													x	0-30 L	-	1	0-19.8	2-26			impinging jet
Yucel & Kakac			dc			x		x	x			x		x								x	3-42 I	14-28	1	0-6			1.25-6.25	varied heater orientation
Drizius, et al.			dc			x	x	x	x			x										x	25-70 L	40-153	3.9-13.4	0-81.3			4-25	swirl flow
Doroschuk, et al.			?			x	x	x	x									x			x		?	225,?	20-167	0-1.2			.75-3	
Ragheb, et al .		x				x		x	x													x	0-28 I	8.3	1	0-1			.034-.1	transition studied
Mel'nikov, et al.			?			x		x	x													x	0-86 E	?	30	0-5.3			2.5-3.5	fouling studied
Chen, et al.			ac			x		x	x			x	x	x	x					x			10-80 I	202,253	1	0-2.6			.1-.4	
Romberg & Harris		x?															x		x				?	?	?	?			?	acoustic meas.
Ragheb & Cheng		x				x		x	x													x	0-28 I	8.3	1	0-1?			.034-.1	
Bennett & Chen			x					x														x	0 I	3	?	.7-3			.16-1.6	pure and binary coolant mixture
Mori, et al.		x	dc																		x		?	75	1	0-1.5	.005-.007			bubble generation
Petukhov, et al.			dc					x	x														0? I	300	1	0-.004			.015-.33	
Koba, et al.			dc			x	x	x	x			x											0-150 I	15-200	1-11	4-21	.5-29.9			small diameter tubes
Tolubinskiy, et al.			dc			x	x	x	x													x	5-80 ?	?	1-5	0-3	.5-2			acoustic measurements
Yilmaz & Westwater					x			x													x		0-5 L	20	1-1.1	0-.75	0-6.7			
Ornatskiy, et al.			x?			x	x	x	x			x						x					9-374 I	25-300	73.6-201	0-6.8			.5-3.5	
Jain, et al.			ac			x	x	x													x	x	10-60 I	124	3.5-4.1	.07-.15			.57-.7	void fraction measurements
Ornatskiy & Sharayevskiy			dc			x	x	x														x	10-120 I	gap=.5mm	1-4	?	1-2.5			acoustic measurements
Fisenko, et al.			x?			x	x		x	x													10-100 I	125	100-180	0-2.9			2	
Leontiev, et al.			ac			x	x	x	x									x					0-145 L	69	68.7-137	0-7.25			.75-2	
Zeigarnik, et al.			ac			x	x	x	x									x					20-160 I	54	5-30	8-62.1			4-25	
Yao			dc					x				x											0-4.5 I	7-136	1-1.1	0-.1	.125-.2		.19-.3	gaps
Styrikovich, et al.			ac			x	x	x	x							x							0-116 L	75,152	68.7-137	0-4.5			1,2	
Ragheb, et al.		x				x		x	x				x							x			0-28 I	D=12 mm	1	0-2.5			.07-.2	ss & transient heating
Ramamohan, et al.			x?			x		x															40-64 A	150	?	.08-2.4	.15-.55			
Jensen & Bergles			dc			x		x	x			x			x							x	0-110 I	83-174	9.4	.05-.8			.57-5.5	straight & coiled flow
France, et al.				x			x	x	x														?	1297	70-153	.5-1.6			.7-3.2	
Agrawal, et al.			ac														x					x	0 I	210	?	.008-.02			.2-.39	plain & swirl flow
Aounallah, et al.			ac														x					x	0 I	48	1-1.4	?			.1,.2	
Del Valle & Kenning			dc			x		x							x						x		20-80 I	21	1	.3-4.7	.8-2			Eddington et al rig
Fukuyama & Hirata			ac			x	x	x	x														?	51	3.5-14.9	0-9			41.5-52.9	questionable mass flux
Katto & Ashida			dc			x	x	x	x														0-10 I	50	19.6-34.3	0-.83			.7-7	
Leung, et al.		x	x			x		x	x													x	0-106 I	1162	97	0-1.25			2-5	different heater configurations
Mishima & Ishii			dc					x	x														?	108	1	?			.005-.035	
Nishikawa, et al.			ac			x	x	x	x														0-24 I	154	34-48	0-.09			.2-1.3	
Papaioannou & Koumoutsos		x						x													x	x	?	-	?	0-.025	0-.1			bubble generation
Rogers, et al.			x?			x		x	x			x		x							x	x	33-93 I	28-54	1.6	.15-2.8			.06-1.2	void fraction measurements
Katto & Yokoya			dc			x	x	x	x			x		x			x						0-16 I	200,333	19.6-34.3	0-.5			1.1-9	
Sekoguchi, et al.			dc			x	x	x				x		x									0-84 ?	91-154	2-16	.12-1.86			.3-2	
Viannay & Karian	x	x				x		x														x	?	?	?	.1-10	.7-11			
Wei & Maa			dc		x			x	x			x				x					x	x	?	10.5-32	?	0-.9	.03-.3			surface tension effect
Ueda, et al.		x				x		x	x			x											x=-.06to.6 I	6,10	2.8	0-.55			.36-1.5	
Sharon, et al.					x	x	x	x						x			x				x		0-15 I	60,100	1.3-7	0-.5			.33,.57	
Bilicki			x?					x	x														0 I	38	6	0-.11			.62-1.2	
Weede & Dhir			dc			x		x	x			x										x	x=-.2to-.3 I	5-26.5	2	0-.84			2.1-6.3	with/without tangntl injection
Klimenko & Sudarchikov			ac			x	x	x															0.2-10 I	185	1.8-7	.01-.04			.17-.75	large length
Kataoka, et al.			dc			x	x	x	x			x								x			0-70 I	26.2-126	1.4-15	0-19.5	.4-4			transient & s.s. heating
Kalayda, et al.			ac				x	x	x	x													0-189 I	.24?	80-160	1-2.55			.5-2	
Baines, et al .		x				x		x	x														5-18 I	-	1	0-1.62	1-5			gravity & jet driven films
Katto & Ohno			dc			x	x	x	x														.1-10 I	100	19.6-34.4	0-.2			.12-2.1	
Kozhelupenko, et al.			x			x	x	x	x					x	x						x		0-80 I	33-50	1-10	.05-1.6	.02-.7			
Hino & Ueda			ac			x		x	x								x				x		10-30 I	40	1.5	0-.2			.51-1.24	fluid temp probe survey

APPENDIX B

Concluded

Investigator	Ref	Yr	Organization	TYPE: Subcool (surf)	TYPE: Sat (bulk)	SURFACE: Tube	Annulus	Rod / Wire	Tube Bundle	Plate, Channel	Ribbon	Matl	COOLANT: Type
Evans, et al.	342	85	Lehigh U		x	x						Inconel 600	water
Edelmam, et al.	241	85	Technion, Isr	x		x						SS	water
Boyd, et al.	97	86	Sandia NL	x		x						copper	water
Fujita, et al.	343	86	Japan		x				x			copper	Freon 113
Fukuyama, et al.	344	86	Japan	x		x						SS	Freon 113
Grigoriev, et al.	345	86	USSR		x	x				x		SS, copper	liq nitrogen
Muller-Steinhagen, et al.	346	86	Canada	x			x	x				SS	heptane
Tolubinskiy & Domashev	235	86	USSR		x?	x?						?	water?
Celata, et al.	128	86	Italy	x		x						SS	R-12
Ueda & Kim	347	86	Japan/Korea	x			x					SS	Freon 113
Aounallah & Kenning	348	87	Oxford U	x		x						cupronickel	water
Yezhova, et al.	349	87	USSR	x		x						?	water
Lee & Shen	350	87	U of Ottawa	x		x	x					SS	water
Inasaka & Nariai	351	87	Japan	x		x						SS	water
El-Genk, et al.	352	88	U of NM	x			x					SS	water
Muller-Steinhagen, et al.	353	88	N Zeal,Can	x			x					SS	H2O,heptane
Yin, et al.	354	88	Can,Taiwan	x		x						Inconel 600	water
Fleishman & Yuen	355	88	U of CA	x?						x		copper	water
Passos & Gentile	356	88	France	x		x						Inconel	Freon-113
Lu & Jia	177	88	China	x		x						thin copper layer	water
Blumenkrantz	357	89	Argentina	x	x				x			?	water
Boyd	358	88	PV A&M U.	x		x						Amzirc®	water
Lee & Simon	359	89	Motorola,UMN	x						x		nichrome,platinum	fluoro FC-72
Celata, et al.	129	89	Italy	x	x	x						SS	R-12
Boyd	174	89	PV A&M U.	x		x						Amzirc	water
Jung, et al.	360	89	UMD,NBS	x	x	x						SS	refrig mixs
Jung, et al.	361	89	UMD,NIST		x	x						SS	refrig mixs
Gu, et al.	110	89	U of KY, APPL	x						x		copper	fluoro FC-72
Polonskii, et al.	362	90	USSR		?	x						SS	water
Hasan, et al.	363	90	AZ State,EPRI	x			x					SS	Freon-113
Cornwell	364	90	UK		x				x			copper	Freon-113
Sivagnanam & Varma	365	90	India	x				x				Pt	var/water
El-Genk, et al.	366	90	U of NM		?				x			SS	water
Shim, et al.	49	90	Canada/Korea	x			x					SS,SS/aluminum	water
Dietrich, et al.	367	90	New Zealand	x	x			x				SS	isobutanol
Vandervort, et al.	368	90	RPI	x		x						SS,Inconel	water
Yuncu, et al.	369	90	U of Miami	x		x						SS, nichrome	Freon-11
Brauer, et al.	370	90	Germany	x			x					SS	R-12
Hosaka, et al.	371	90	Japan	x		x						SS	Freon-113
Hihara & Saito	372	90	Japan		x	x						SS	R22,R114,mix
Hahne, et al.	373	90	Germany	x		x						copper	R-12
Dougherty, et al.	374	90	Columbia U.	x		x						SS,Inconel	water
Liu & Kakac	375	91	U of Miami	x		x						nichrome	R-11
Ravindran, et al.	376	91	India	x	x		x					SS	water
Boyd	377	91	PV A&M U.	x		x						copper	water
Feng & Johannsen	378	91	Germany	x		x						Monel	water
Celata, et al.	379	91	Italy	x	x	x						SS	R-12 -
Kocamustafaogullari & Wang	380	91	U of WI			x						-	water
Hasan, Roy & Kalra	381	91	AZ St U/EPRI	x			x					SS	R-113
Balino & Converti	382	91	Argentina		x		x					SS	Freon-12
Akhanda & James	51	91	Bangladesh/UK	x						x		SS	water
Boyd	383	91	PV A&M U.	x		x						zirconium-copper	water®
Gyulmamedov, et al.	384	91	USSR	x	x	x						SS	water/TrilonB
Galloway & Mudawar	385	92	Purdue	x	x					x		copper	fluoro FC-72
Dykas & Jensen	386	92	RPI		x				x	x		SS	Freon-113
Leroux & Jensen	387	92	RPI	x	x				x	x		SS	Freon-113
Celata, et al.	132	92	Italy	x	x	x						SS	R-12
Orozco & Chen	388	92	U of IL	?						sphere		copper	Freon-113
Sadasivan & Lienhard	60	92	U of Houston	x	x			x				SS, Nichrome	var,water
Panchal, et al.	389	92	ArgonneNL,UIL	x	x	x						aluminum	R-11
Willingham & Mudawar	390	92	Purdue	x						x		electronic chip	FC-72
Vandervort, et al.	391	92	RPI	x		x						SS	water
Vandervort, et al.	80	92	RPI	x		x						SS,Ni,brass,Inconel	water

Investigator	COOLANT Treatment [1]: dis	dei	dem	deg	gas	untr	nwtd	Flow Orientation	HEATING: Coal,Gas Fired	Elect Htr	Elect Resis	Liquid Metal	Steam, Fluid	EFFECTS STUDIED: dTsub [2]	Press	Vel [3]	Burnout crit pt	Gas Contnt	Accel	L/D [4]	Wall Matl	Dh [5]	Wall Thk
Evans, et al.							x	vert			dc				x	x	x						
Edelmam, et al.	x							vert			dc			x		x							x
Boyd, et al.		x		x				horiz			x					x	x						
Fujita, et al.								horiz		x					x								
Fukuyama, et al.								horiz			ac			x	x	x	x						
Grigoriev, et al.								horiz,vert			?				x	x					x		
Muller-Steinhagen, et al.								horiz,vert		x				x	x	x							
Tolubinskiy & Domashev							x	vert?		x?							x						
Celata, et al.								vert			x				x	x	x						
Ueda & Kim								vert			ac			x		x	x						
Aounallah & Kenning							x	vert			dc			x		x							
Yezhova, et al.				x	x			not defined			x			x	x	x	x	x					
Lee & Shen			x					not defined			?					x	x						
Inasaka & Nariai		x						vert			dc			x		x	x			x		x	
El-Genk, et al.		x						vert			dc			x		x	x					x	
Muller-Steinhagen, et al.				x	x			vert		x				x	x	x		x					
Yin, et al.		x						vert			dc			x	x		x						
Fleishman & Yuen							x	horiz?	x							x							
Passos & Gentile								vert			x?			x	x	x	x						
Lu & Jia	x		x	x				horiz			x				x	x							
Blumenkrantz							x	?			x			x	x	x	x						
Boyd		x		x				horiz			dc					x	x						
Lee & Simon								horiz			dc			x	x	x	x			x			
Celata, et al.								vert			x			x	x	x	x			x			
Boyd		x		x				horiz			dc					x	x						
Jung, et al.								horiz			dc			x		x							
Jung, et al.								horiz			dc					x							
Gu, et al.								horiz		x				x		x	x		x				
Polonskii, et al.			x					horiz			x?				x	x							
Hasan, et al.								vert			dc			x	x	x							
Cornwell								vert		x													
Sivagnanam & Varma								vert			dc			x		x							
El-Genk, et al.			x					vert			dc					x							
Shim, et al.				x				vert			x?			x	x	x	x						
Dietrich, et al.								vert			dc			x		x				x			
Vandervort, et al.	x		x	x				vert?			dc			x	x	x	x			x		x	
Yuncu, et al.								vert			x					x							
Brauer, et al.								vert			dc			x	x	x							
Hosaka, et al.								vert			ac			x	x	x	x					x	
Hihara & Saito								horiz			dc					x							
Hahne, et al.								vert		x				x	x	x							
Dougherty, et al.		x		x				vert			dc			x	x	x				x		x	
Liu & Kakac								vert			dc			x		x							
Ravindran, et al.							x	horiz			ac			x	x	x						x	
Boyd		x		x				horiz			dc					x	x			x			
Feng & Johannsen	x		x	x				vert		x				x	x	x	x						
Celata, et al.								vert			x			x	x	x	x			x			
Kocamustafaogullari & Wang							x	horiz								x							
Hasan, Roy & Kalra								vert			dc			x	x	x							
Balino & Converti								horiz			dc					x	x						
Akhanda & James	x	x		x				horiz			ac			x		x							
Boyd		x		x				horiz			dc					x	x						
Gyulmamedov, et al.	x			x				vert			ac			x	x	x	x						
Galloway & Mudawar				x				circumfr		x				x	x	x	x		x	x			
Dykas & Jensen				x				vert		x	dc				x	x	x						
Leroux & Jensen				x				vert			dc				x	x	x						
Celata, et al.								vert			x			x	x	x	x			x			
Orozco & Chen								vert	x					x		x	x						
Sadasivan & Lienhard							x	vert			ac,dc					x	x		x				
Panchal, et al.								vert					x										
Willingham & Mudawar				x				horiz		x				x		x	x					x	
Vandervort, et al.		x		x				vert?			dc			x	x	x	x			x		x	
Vandervort, et al.		x		x				vert			dc			x	x	x	x	x		x	x	x	

Investigator	Rough-ness	Press Drop	Uni/Nonun Ht Flux	Stability	Transient	Optical	Other	PARAMETER RANGE: dTsub deg-C [2]	L/D [4]	Press / bar	qdot [6] MW/sq-m	Vel m/s	mdot [7] kg/s	G [8] Mg/m2 s	Comments
Evans, et al.								0 I	88	2.4-5.7	.02-.06			.013-.085	vapor probe measurements
Edelmam, et al.							x	78-83 I	167,169	1	.03-.24			.08-.25	void fraction measurements
Boyd, et al.								186 I	116	16	1.5-4.25			.63-3.5	
Fujita, et al.	x						x	0 I	4.8	1-5	0-.1			?	
Fukuyama, et al.								59-140 L	50	5.8-23.8	0-10.1			28-56	questionable mass flux
Grigoriev, et al.	x							?	?	1-20	.006-.15				
Muller-Steinhagen, et al.							x	92-139 L	7,625	1-7.9	0-.4			.005-.7	heater geometry varied
Tolubinskiy & Domashev							x	?	100	98-147	0-5.5			1	CHF detector
Celata, et al.					x			23 I	307	12.5-27.5	.03-.085			.4-1.6	flow transients
Ueda & Kim								10-30 I	40	1.5	0-.45			.26-1.2	
Aounallah & Kenning								5-15 ?	52	1.1	0-.17			.1-.2	
Yezhova, et al.								0-85 ?	D=8 mm	98.1-167	0-5.3			1-3.7	
Lee & Shen	x							?	gap=12 mm	?	0-1.5			.05-.2	falling films also
Inasaka & Nariai		x						0-75 ?	3-100	1	?			7-20	
El-Genk, et al.								44-75 I	39-68	1.2	.16-1.4			0-.26	
Muller-Steinhagen, et al.								32-151? L	7	1-7.3	0-.37			.2-.76	
Yin, et al.								10-80 I	273	10-214	0-2			2	
Fleishman & Yuen		x					x	?	?	?	.15-.6		.0018-.12		radial flow thru wire mesh
Passos & Gentile								5-40 I	6.25	1.5-4	0-.6			.4-1.3	transition evaluation
Lu & Jia		x					x	5-12? I	72.3	1.8-2.4	.058-.077			.82-1.29	void fraction measurement
Blumenkrantz								0-160 I	232	68-151	?			.94-6.65	
Boyd		x						0-74 E	96.6	7.7	0-42			4.4-41	
Lee & Simon	x							13-46 I	.05-.65	1.3-2.8	0-1.7	2.5-8.5			
Celata, et al.					x			0-23 I	153-299	12.5-27.5	0-.12			1-1.5	transient & s.s. heating
Boyd		x						183 E	96.6	16.6	0-42			4.4-32	
Jung, et al.								upto x*.95 E	444	2.6-4	.01-.045		.016-.045	.25-.72	
Jung, et al.								up to x*.9 E	444	pr=0.08	.01-.045		.016-.045	.25-.72	follow-on to previous expt
Gu, et al.								.5-38 L	1	1	0-1.4	1-4			straight & curved flow
Polonskii, et al.			x					?	?	100-140	.2-1.3			.5-2	
Hasan, et al.								20-37 I	121	2.4-3.1	.01-.15			.58-1.1	
Cornwell							x	0 L	?	1	0-.039			.1	
Sivagnanam & Varma								10-40 A	1650	?	.2-3	.15-1			binary mixtures
El-Genk, et al.							x	?	71	?	?		.013-1.52		
Shim, et al.	x							42-120 I	75-94	1.1-3.5	0-11.3			1-5.9	finned & unfinned geometries
Dietrich, et al.							x	?	-	1	0-.5	.01-.4			coiled wires
Vandervort, et al.				x				40-135 E	2-25	6, 12	10-100			5-40	many premature failures
Yuncu, et al.		x		x			x	?	81-86?	1	0-.028		0-.035		surface enhancements
Brauer, et al.		x					x	10-75 I	46.4	10-33	0-.12			.5-3	void fraction measurement
Hosaka, et al.								50-80 ?	50	11-24	0-6			9.3-32	
Hihara & Saito								0 I	500	6	.008-.026			.2-.35	
Hahne, et al.								2.2-10.8 I	25	7-8.1	0-.045			.2-2.25	
Dougherty, et al.		x		x				60-110 I	85-158	1.38,3.45	1.26-3.16		0-2.5		
Liu & Kakac				x				?	80.7	?	.022-.055		.007-.02		
Ravindran, et al.								4-10 I	27-92	1.1, 1.5	.0085-.034			.017-.38	
Boyd		x						129 I	49	4.5	0-11			1.56-8.55	
Feng & Johannsen							x	5-57 I	5.2	1.1-12	.02-3.3			.01-1	transition boiling
Celata, et al.		x			x			0-23 I	153-299	12.5-27.5	0-.12			1-1.5	trans & s.s. heating,press,flow
Kocamustafaogullari & Wang							x	?	306	?	-	3.75-5.7			void fract meas, unheated tube
Hasan, Roy & Kalra						x	x	20-37 I	124	1.8-2.5	.04-.14			.58-.8	vapor characteristics measured
Balino & Converti							x	0 I	23.4	10.3	0-.32			.24-1.6	eccentricity effects
Akhanda & James	x						x	5-30 I	13.8	1	0-.55	.2-1.4			finned & unfinned geometries
Boyd								188 I	96.6	16.6	0-36			4.4-31.5	
Gyulmamedov, et al.							x	x=-.7to.7 I	150	8-160	0-2.4?			.1-1.5	chelating agent effect
Galloway & Mudawar							x	35-50 L	1-4	1-1.7	0-1.65	?			centrifugal force effect
Dykas & Jensen		x						x=-.04to.7 L	10.4	1.5, 5	0-.33			.05-.5	
Leroux & Jensen		x						0-8 I	10.4	1.5, 5	0-.33			.05-.5	
Celata, et al.					x			0-23 I	153,298	12-27.7	.026-.12			.4-1.6	trans & s.s. power,press,flow
Orozco & Chen	x				x	x	x	?	-	1	0-.55	1			transient incl. film boiling
Sadasivan & Lienhard						x	x	0-5 L	33-267	?	0-7.5	.1-3			acceleration, wettability effct
Panchal, et al.							x	?	272	?	?	?			spirally fluted & plain tubes
Willingham & Mudawar						x		3-36 I	3.6-13.3	1.36	0-.8	.13-4			flow over electronic chips
Vandervort, et al.		x						0-150 E	1-100	1-22	0-104			5-50	
Vandervort, et al.		x				x	x	10.5-142 E	1.66-49.	1.31-22.8	0-123.8			3-42.7	void fraction measurement

NOTES:

[1] coolant treatment
- dis distilled
- dei deionized
- dem demineralized
- deg degassed
- gas gassed
- untr untreated
- nwtd no water treatment discussed

[2] dTsub subcooling, Tsat-Tbulk
I inlet, L local, E exit, A average (of inlet & outlet), ? unknown location and/or value
(in some cases listed as inlet, the inlet bulk temp & exit pressure were used to compute dTsub)

[3] Vel velocity

[4] L/D for tube D = Dh
for circular annulus D = Dh = 2(a-b) a = outer radius, b = inner radius
for rect duct or channel D = Dh = 2ab/(a+b) a = width, b = height
for ribbon D = W, W = width

[5] Dh hydraulic diameter (see note 4)

[6] qdot heat flux

[7] mdot mass flux

[8] G mass velocity

[9] where a question mark directly follows an item or value it is uncertain but probable

NOMENCLATURE

A	Surface area, ft^2
a	Centrifugal acceleration, ft/hr^2
Bo	Boiling number, $= q''/G\ i_{fg}$
c, c_p	Specific heat, Btu/lbm °F
C_D	Drag coefficient for a sphere
CHF	Critical heat flux, Btu/ft^2 hr
D	Diameter, ft
d	Bubble departure diameter, in.
DNB	Departure from nucleate boiling
erfc	Complementary error function
f	Friction factor as given in Eq. (5)
G	Mass velocity, $= \varrho V$ except as defined in Eq. (12), lbm/ft^2 hr
g	Acceleration of gravity, ft/hr^2
g_c	Gravitational constant, ft/hr^2
h	Heat-transfer coefficient, Btu/ft^2 hr °F
i	Enthalpy, Btu/lbm
i_{fg}	Latent heat of vaporization, Btu/lbm
J	Energy conversion factor, = 778 ft-lbf/Btu
k	Thermal conductivity, Btu/ft hr °F

L	Length, ft
m_v	Vapor rate, lbm/hr
Nu	Nusselt number, = hD/k
p	Pressure, psia
Pr	Prandtl number, = $\mu c/k$
$q'',q/A$	Heat flux, Btu/ft^2 hr, except as noted
q''/V	Internal heat generation rate per unit volume
R	Ideal gas constant
r	Characteristic radius, ft
R'	Laplace number, = $r[g(\varrho_\ell - \varrho_v)/\sigma]^{1/2}$
Re	Reynolds number, = $\varrho VD/\mu$
T	Temperature, °F
t	Wall thickness (Fig. 35)
$t_{\dot{m}/2}$	Time to reach half of initial flow, sec
t_{tr}	Flow rate transit time, sec
V	Velocity, ft/sec
x	Vapor quality
Z	Length, as defined in Figs. 21 and 24

SYMBOLS

α	Thermal diffusivity, ft^2/hr
ΔC	Excess concentration of the volatile component in the vapor, percent (see Fig. 17)

Δi_i	Enthalpy difference due to inlet subcooling, $= i_{local} - i_{in}$, Btu/lbm
ΔT_{bulk}	Complete temperature difference, $= T_w - T_b$, °F
ΔT_{sat}	Superheat, $= T_w - T_{sat}$, °F
ΔT_{sub}	Subcooling, $= T_{sat} - T_b$, °F (may be inlet, outlet, local, or average)
μ	Viscosity, lbm/ft hr
ϱ	Density, lbm/ft^3
σ	Surface tension, lbf/ft
τ	Shear stress, lbf/ft^2 [except as defined in Eqs. (38) and (39)]
ϕ_m	Specific mass flow rate in Eq. (39), kg/m^2 sec

SUBSCRIPTS

avg	Average
B	Fully developed nucleate boiling
Bi	Extension of the fully developed nucleate boiling curve to incipience point
b	Bulk fluid
CHF	Critical heat flux conditions
conv	Pure convection conditions
cr	Critical heat flux conditions
D	Diameter
D_e	Equivalent or hydraulic diameter
e	Equivalent or hydraulic

f	Film
FC	Forced convection
h	Hydraulic
htr	Heater
I	Inner
i,in	Inlet
ℓ	Liquid
LB	Subcooled boiling incipience
m	Mean
O	Outer
o	Outlet
r	Reduced
sat	Saturation
ss	Steady-state conditions
tr	Transient conditions
ts	Test section
v	Vapor
w	Wall