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PHASE CHANGE HEAT TRANSFER – A REVIEW OF PURDUE RESEARCH

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ABSTRACT

Research carried out a Purdue University in the area of phase change heat transfer (e.g., boiling) has been briefly reviewed in this paper. The hallmark of Purdue research is that basic studies were motivated by industrial applications and the results of basic studies culminated in solutions to practical problems. The reviewed research work is divided into three distinct areas: material processing, electronic cooling, and phenomenological studies of critical heat flux with applications in the power industry.

INTRODUCTION

This review is focused on the investigations carried out in the area of boiling heat transfer by Professors F.P. Incropera, I. Mudawar, S. Ramadhyani, and R. Viskanta and their students in the School of Mechanical Engineering at Purdue University. The review is divided into three broad areas: material processing, electronic cooling, and phenomenological studies of critical heat flux with applications in the power industry. These faculty and students deserve to be applauded for their ability to perform basic research while not losing sight of the intended applications. Key results obtained in each of the areas are described in the following.

NOMENCLATURE

$C_1 \ldots C_5$	empirical constants
D	tube inside diameter
G	mass velocity
$h_{f,0}$	saturated liquid enthalpy at outlet
$h_{fg,0}$	latent heat of vaporization at outlet pressure
h_i	enthalpy of liquid at inlet
L	heat length of tube
q_{CHF}	critical heat flux
ρ_f	liquid density
ρ_g	vapor density
σ	interfacial tension

<u>Materials Processing</u>

The product quality from many manufacturing processes depends on thermal response of the material to the cooling methods that are employed during manufacturing. The product quality not only includes physical dimensions, but also the mechanical properties. Some examples of manufacturing processes of interest are hot rolling, casting, extrusions, forging, and annealing.

Zumbrunnen, Viskanta and Incropera [1] investigated the effect of surface motion on forced convection film boiling. This situation arises during cooling by planar jets of a hot plastic steel strip after rolling and during continuous annealing of a cold rolled strip. Figure 1 shows various heat transfer regimes on a steel strip with film boiling being dominant over a larger portion of the plate. It was shown through two-layer integral analysis for laminar flow that with plate motion in the same direction as the subcooled liquid, the vapor film thinned, thereby improving the heat transfer. However, when the plate motion was opposite to the liquid flow direction, the vapor film thickened with a corresponding reduction in heat transfer. Subsequently, similarity solutions were obtained for laminar film boiling and analyses were extended to turbulent film boiling.

Filipovic, Incropera, and Viskanta [2] have investigated, experimentally and theoretically, quenching phenomena associated with a water wall jet as encountered during cooling of a rolled strip or during continuous casting. A quench front (just downstream of the region II in Fig. 1) forms and it travels in the direction of liquid flow. Upstream of the quench front, forced convection and nucleate boiling exist, whereas film boiling prevails downstream. The quench front region, as

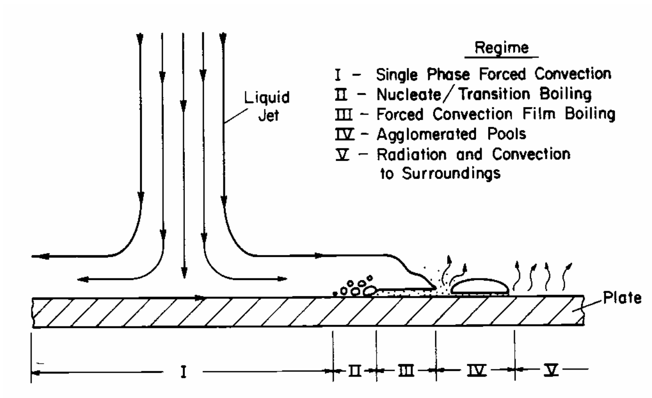

Figure 1: Boiling regimes during quenching of steel strip with a planar jet.

shown in Fig. 2, is marked by the existence of transition boiling. It was noted that the length of this region could be as much as 20 mm. The upstream edge of the region marked the critical heat flux condition, whereas the downstream edge, the wetting temperature. At the wetting temperature, liquid-solid contact was established. An apparent quenching temperature was defined at which, as a result of axial conduction in the solid, cooling began to take place at a rate faster than in film boiling. Subsequently, the authors developed an accurate method for quantitatively determining quenching and apparent quenching temperatures from temperature-time plots. It was shown that rewetting temperature and quench front velocity depended on temperature of the solid prior to quenching, liquid subcooling, and flow velocity.

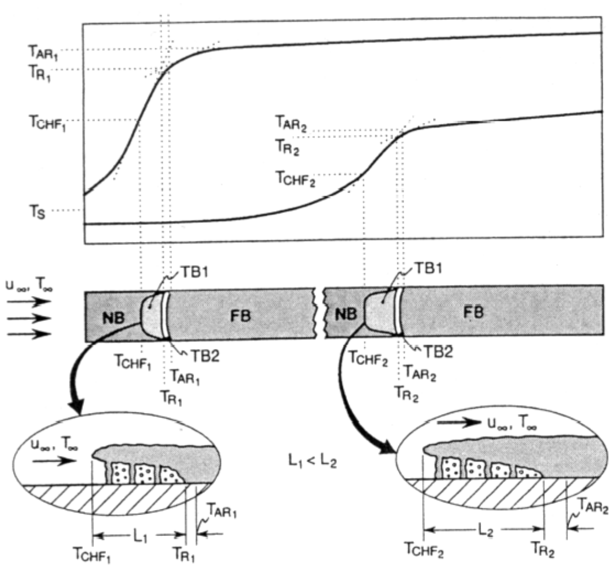

Figure 2: Details of quench front region.

Hall, Incropera, and Viskanta [3] studied the effect of entrainment of gas on quenching characteristics of a planar jet of subcooled water. They found that with gas entrained in the subcooled liquid, heat transfer could be reduced in film boiling, thereby reducing the deformation of a billet during casting. The presence of air also reduced the temperature excursion that occurred after cessation of boiling during quenching by reducing the wall superheat for transition from nucleate boiling to single phase convection. Wolf, Incropera and Viskanta [4] have also experimentally investigated local heat transfer as opposed to average heater transfer during impingement of a planar jet of liquid. They found that jet velocity had influence only on single phase and partial nucleate boiling. The extent of partial nucleate boiling (range of wall superheat) depended on the distance from the stagnation point (decreased with decrease in distance). Single phase heat transfer coefficient was only sensitive to locations near the stagnation point.

Mudawar and co-workers have focused on the use of sprays to quench aluminum products post extrusion, forging, or during continuous casting. A very rapid quenching can lead to plastic deformation which, in turn, can cause warping of the product. On the other hand, slow quenching can lead to undesirable metallurgical properties and degradation of mechanical strength. In their pursuit to develop an ideal quenching rate that avoided the two extremes of warping and low mechanical properties, Mudawar and Valentine [5] and Klinzing, Rizzi, and Mudawar [6] took the first logical step of developing correlations for film boiling heat transfer, minimum film boiling temperature, minimum film boiling heat flux, transition boiling heat transfer, maximum heat flux, heat transfer in nucleate boiling, and temperature for the onset of single phase cooling and single phase heat transfer during spray cooling. A key feature of these correlations as that they were dependent on local spray properties such was Sauter mean droplet diameter, droplet velocity, droplet volumetric flux, and liquid temperature as opposed to properties that are defined at the nozzle exit. Deiters and Mudawar [7] used this information along with a solution of two dimensional transient conduction in the solid, to develop a scheme whereby sprays of desired strengths could be used on different sections of a product. Figure 3 shows their proposed scheme. When heavy density sprays were used on all surfaces a large temperature difference (325°C maximum) could occur at different locations in the solid during quenching. However, when heavier spays were used to quench thicker sections and lighter sprays for thinner sections, the temperature differences were substantially reduced ($\simeq$ 75°C maximum). Figures 3a and 3b show the temperature histories at different locations on the solid.

Subsequently, Hall and Mudawar [8] used the quenching rate information to predict the mechanical strength of parts made of aluminum alloys. Through similar studies they developed smart quenching systems that could select spray nozzle configurations and other flow parameters to obtain superior mechanical properties for parts made of aluminum.

Microelectronic Cooling

With rapid increase in chip switching frequency, and in turn, power densities, we have almost reached the limit where the direct cooling of chips with liquid must be considered. The Purdue Heat Transfer group has been a leader and visionary in this regard, as it started to investigate the liquid cooling of simulated chips almost fifteen years ago. Various cooling techniques that have been considered are pool boiling, forced flow boiling in micro and mini channels, jet impingement cooling, falling liquid films, and spray cooling.

Anderson and Mudawar [9] experimentally studied the effect of artificial cavities and various surface enhancement schemes on hysteresis in the boiling curve and on nucleate and

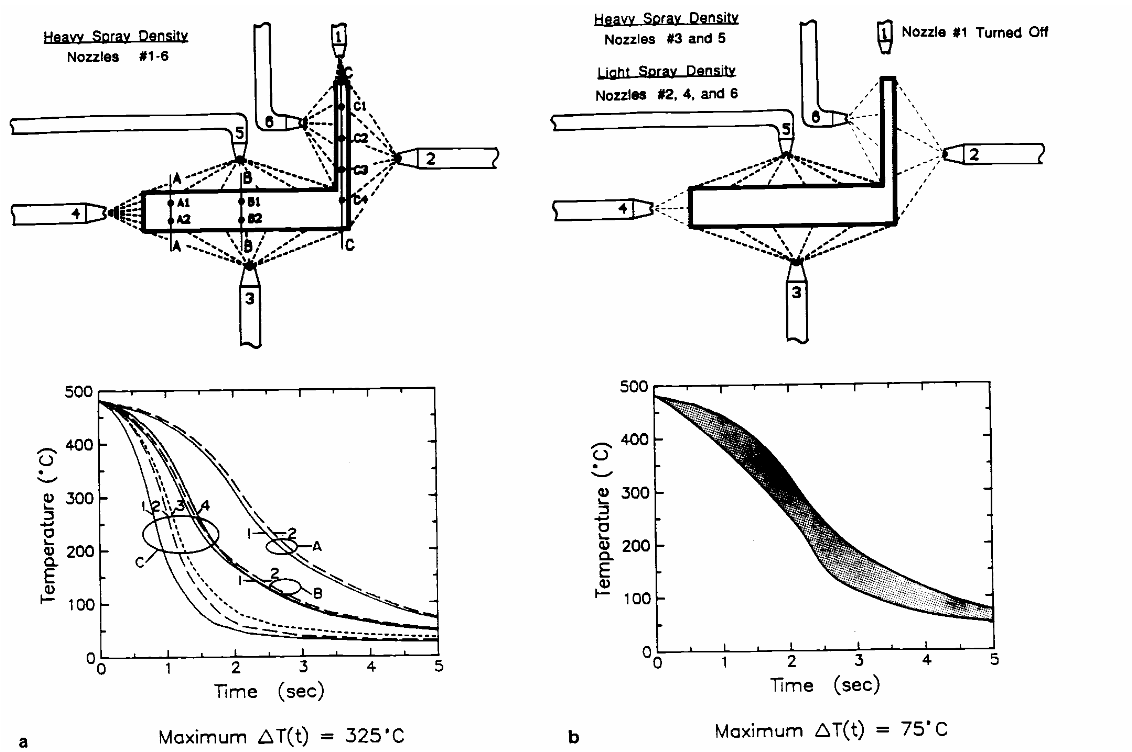

Figure 3: Variations with time of temperatures in the solid during quenching under unoptimized and optimized conditions.

critical heat fluxes using a simulated chip placed in a pool of dielectric liquid, FC-72. They found that temperature overshoot increased with increase in the waiting period between various experiments. Artificial cavities with a diameter of about 300 μm had little effect on temperature overshoot and on nucleate boiling heat flux. Microfins, microstuds, and microgrooves tended to shift the nucleate boiling curve to the left and increase the maximum heat flux based on a projected area of the chip. The effect of preceding chips placed in an array was to increase the thickness of the void layer on the succeeding chips and thereby increase the wall superheat for a given heat flux. Liquid subcooling tended to reduce the thickness of the bubble boundary layer and increase the maximum heat flux. An optimum (with highest CHF) enhanced surface was developed by placing a single cylindrical stud on the chip and forming microstuds, radial microgrooves, or axial microgrooves on the studs and finally roughening the surface by blasting it with silica, water, and air slurry. On the optimum surface, critical heat fluxes of more than 150 W/cm^2 were obtained, when the liquid had a subcooling of 35°C.

Mudawar and Maddox [10] experimentally investigated critical heat flux on a 12.7 × 12.7 mm^2 surface simulating a chip surface placed flush with one of the walls of a vertical rectangular channel having a cross-section of 12.7 × 38.1 mm^2. Using FC-72 as the test liquid, they found that critical heat flux increased with subcooling and velocity. They identified different CHF mechanisms for high and low velocities and developed a correlation applicable for all ranges of velocity. Subsequently, they studied the effect of surface enhancement schemes such as pin fins, microstuds and microgrooves on flow boiling CHF. They found all the methods enhanced CHF over a plane surface, but pin fins performed the best. Studies of boiling and critical heat when a number of simulated electronic chips were placed in the direction of flow showed that the last chip (#9) was the first to develop nucleate boiling, followed by the upstream chips in order. A slight decrease in CHF in the downstream direction was observed as the vapor boundary layer thickness increased. The CHF data from all of the chips was found to be within ±12%, which was comparable to experimental uncertainty. Experiments with different channel heights (2, 5 and 10 mm) showed that the temperature overshoot was the highest for the 2 mm high channel. Maximum values of critical heat flux for equal flow velocities were obtained for the 5 mm high channel. It was postulated that it was for this height that bubble removal process from the heater surface was most effective while liquid subcooling persisted throughout the channel.

Heindel, Ramadhyani, and Incropera [11] investigated the cooling of simulated chips placed on the lower wall of a horizontal rectangular channel and protruding into the flow. Ten discrete inline heat sources were used with FC-72 as the test liquid. Temperature overshoot was observed consistent with earlier studies. However, increasing the flow velocity was found to reduce the temperature overshoot. Subcooling of the liquid also reduced the magnitude of the temperature overshoot. A reduction in the height of the channel restricted vapor escape from the heater surfaces, especially those in the downstream direction. As a result, the nucleate boiling heat flux on these heaters, at a given wall superheat, was lower than that in the upstream heaters.

Bowers and Mudawar [12] experimentally investigated the concept of placing mini (D = 2.54 mm) and micro (D = 510 μm) channels in the substrate supporting a chip. The effective heated length of the channels was 1 cm and R-113 was used as the test liquid. In both cases a number of parallel channels formed in the copper substrate and connected to common manifolds were operated. No temperature overshoot was reported at onset of flow boiling in the channels. In both mini and micro channels critical heat fluxes higher than 200 W/cm^2 were obtained under a modest pressure drop. Because of spreading of heat across channels, no significant rise in temperature after CHF occurred. As a result, at low flow rates, almost all of the liquid could be evaporated without experiencing large excursion in temperature. Boiling was found to accommodate higher heat fluxes than single phase flow under similar pressure drop conditions. However, a mini channel was preferred because of lower pressure dorp and reduced propensity for blockage. In a subsequent work, Bowers and Mudawar developed design criteria for channel pitch and thickness of the substrate while taking into consideration, the constraints imposed by thermal diffusion. They also developed correlations to two phase pressure drop so that optimum configurations under constraints of pressure drop and flow rate could be determined.

Mudawar and Wadsworth [13] investigated the use of a confined rectangular jet to cool a simulated 12.7 × 12.7 mm^2 chip with FC-72. Two regimes for critical heat flux (CHF) were identified. At medium velocities, the critical heat flux increased with flow velocity, whereas at high velocities, the critical heat flux was either insensitive or decreased with velocity. In the medium velocity regime, the critical heat flux increased with liquid subcooling and nozzle width, but was insensitive to channel height. The reduction in CHF at high velocities was postulated to be caused by reduced liquid subcooling in the streamwise direction. Critical heat fluxes as high as 250 W/cm^2 were obtained. A correlation for critical heat flux in terms of fluid properties, jet velocity, nozzle width, and liquid subcooling was developed for the medium velocity regime.

Spray cooling of a simulated electronic chip with full cone sprays of FC-72 and FC-87 has been experimentally investigated by Mudawar and Estes [14]. It was found that the nozzle to surface distance played an important part in determining the magnitude of critical heat flux, which increased with volumetric flux and liquid subcooling. Critical heat fluxes higher than 100 W/cm^2 were obtained. A correlation of critical heat flux in terms of volumetric flow rate, liquid subcooling, and droplet Sauter mean diameter was developed.

Mudawar *et al* [15] investigated cooling of vertically mounted heat sources with a liquid film falling over the surface. Vigorous nucleate boiling was found to disrupt the film and break it into liquid ligaments and droplets. Near critical heat flux condition, liquid was observed to separate from the heater surface and the critical heat flux occurred when thin liquid film left on the surface dried out. Critical heat flux was found to increase with reduction in heater length and increase in film velocity. Onset of the CHF condition was attributed to Helmholtz instability which led to separation of the flow, while leaving a thin liquid film on the surface.

Phenomenology and Correlations for Critical Heat Flux in Flow Boiling

Professor Mudawar and his co-workers have conducted a number of fundamental experimental studies to develop a phenomenological understanding of critical heat flux during flow boiling. They have used this understanding to develop mechanistic models/correlations for subcooled critical heat flux. They have also made a significant contribution to the technical literature by collecting and assessing more than 32,000 critical heat flux data points available in the literature.

Galloway and Mudawar [16] carried out photographic observations of the boiling phenomena, pre- and post-CHF, on a 1.6 × 12.7 mm^2 heater, mounted flush on the smaller side of a 1.6 × 6.4 mm^2 vertical channel. The experiments were conducted with FC-87 as the test liquid. Visual observations showed that, at low heat fluxes, discrete bubbles were formed on the heater surface. At low velocities, nearly hemispherical bubbles lifted off normally to the heater surface. However, at higher velocities, bubbles were elongated and slid along the heater surface before lift off. A liquid film of a thickness of about 60 μm was observed to be trapped between the sliding bubble and the heater surface. With increase in heat flux to about 60% of the value at critical heat fluxes, bubbles merged into large vapor masses which appear to acquire a wavy interface with a well defined wavelength. A thin liquid layer was deposited on the heater when wave troughs touched the heater surface. The regions over which this thin layer of liquid was deposited on the surface shrank with increase in heat flux. Heat was removed from the thin film by evaporation or nucleate boiling in the film. Critical heat flux condition was proposed to occur when the vapor effusion rate from the interface at the wetting front was able to lift the interface, thereby disrupting the liquid supply to the heater. Photographs in Fig. 4 show the wetting front and the interface lift off when the imposed heat flux slightly exceeded the critical heat flux. The lift off fronts progressed in the downstream direction. The dominant wavelength in the vapor liquid interface and its amplitude were found to decrease nonlinearly with flow velocity. In casting the visual observations into a mechanistic model, the two dimensional wavelength was determined from the Kelvin-Helmholtz instability considerations. The maximum heat flux condition was established when the momentum flux of the vapor leaving the interface was able to overcome the pressure difference that existed across the concave interface at the location where liquid-solid contact was established. The mean thickness of the vapor film and superficial velocities of vapor and liquid that were needed to evaluate the pressure difference at the onset of interfacial instability was determined by invoking conservation of mass and momentum across the vapor and liquid streams. Predictions from the model were found to be in very good agreement with critical heat flux data.

Subsequently, Sturgis and Mudawar [17] extended both the visual observations and the modeling of critical heat flux on longer upward facing horizontal surfaces ($\simeq$ 102 mm in length). Visual observations similar to those reported by Galloway and Mudawar were made except that wavelength, the amplitude, and liquid contact length were found to increase along the flow direction, but decrease with liquid velocity and subcooling. The ratio of the length of the region wetted by the liquid and that occupied by the vapor was found to remain constant in the flow direction for a given subcooling. Photographs of vapor

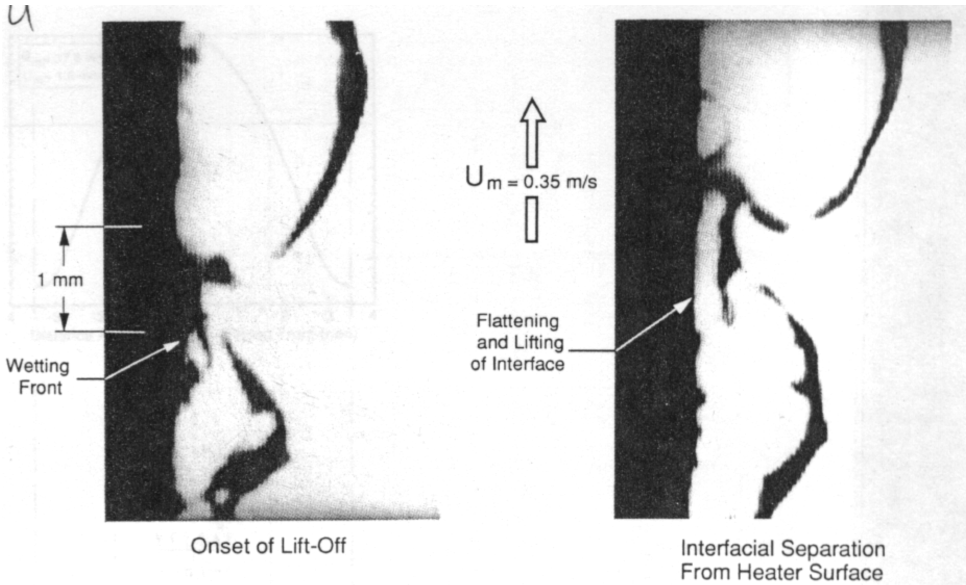

Figure 4: Photographs of wetting front and interface lift off.

structure, as given in Fig. 5, revealed existence of both wave like and overhanging configurations. The most prominent configuration was the wavelike. The overhanging structures were seen only at higher subcoolings. A model similar to that developed earlier for the short length vertical surface was developed for the longer surface. This model reflected improvement over the previous study of Galloway and Mudawar in that an analytical expression was developed for the heat flux at the interface lift off. The model was validated with saturated low subcooling data.

To understand, explicitly, the effect of heater length on CHF, Gersey and Mudawar [18] continued their photographic and model studies on horizontal plates of 10, 30, and 110 mm in length, flush mounted along one wall of a rectangular channel. They also investigated the effect of plate angle from horizontal to inclined at 45° to vertical. It was found that a mechanism similar to that for shorter plates was present for longer plates, except that the distance between wetting fronts or wave troughs increased in the flow direction. This distance was as much as four times the critical Helmholtz wavelength in the middle of the longest plate and remained constant further downstream. This, in turn, led to a smaller number of wetting fronts on the plate and reduced critical heat flux. The critical heat flux was triggered with the lifting of the most upstream wetting front which was followed by the downstream fronts. No appreciable effect of surface orientation on the phenomena was found. The earlier model of Galloway and Mudawar, when modified to include the reduction in number of wetting fronts, was found to be compare well with their data.

Mudawar and Bowers [19] experimentally investigated critical heat flux in short, small diameter tubes using highly subcooled water. In the experiments, steel tubes with an inside diameter varying 0.406 – 2.54 mm, and length to diameter ratio of 2.4 to 34.1 were used. The tubes were heated by passing D.C. current through them. Water temperature at the inlet was varied from 18 – 70°C with mass velocities varying from 5,000 – 134,000 kg/m^{2}s. The outlet pressures investigated were 2.5 to 172.4 bars. The critical heat flux was found to increase with mass velocity, liquid subcooling, and decrease in tube diameter and decrease in heated length to diameter ratio. The critical

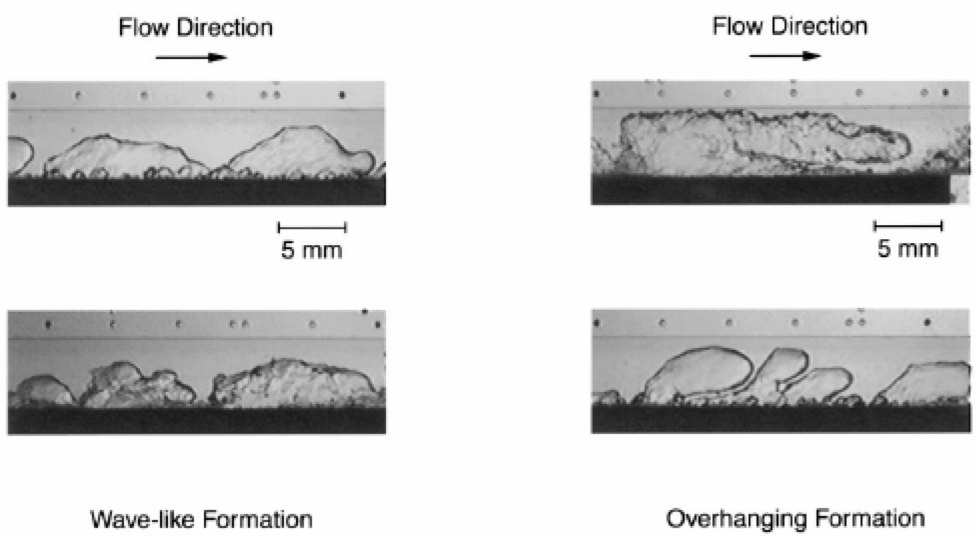

Figure 5: Observed vapor structures prior to CHF under subcooled flow boiling conditions.

heat flux initially decreased with pressure up to 30 bars and, thereafter, remained constant with pressure until it decreased as critical pressure was approached. Because of high pressure drops encountered, it was suggested that a proper account for local pressure be made while developing correlations or models. The highest ever reported critical heat flux of 27.6 kW/cm^2 was obtained in a 0.406 mm diameter tube with a length of 5.8 mm at a mass velocity of 120,000 kg/m^{2}s and water inlet temperature and pressure of 27°C and 160 bars, respectively. A correlation [20] based on the inlet conditions and employing five adjustable constants was proposed,

$$\frac{q_{CHF}}{G h_{fg}} = \frac{C_1\left(\frac{G^2 D}{\rho_f \sigma}\right)^{C_2}\left(\frac{\rho_f}{\rho_g}\right)^{C_3}\left[1 - C_4\left(\frac{\rho_f}{\rho_g}\right)^{C_5}\left(\frac{h_i - h_{f,0}}{h_{fg,0}}\right)\right]}{1 + 4C_1C_4\left(\frac{G^2 D}{\rho_f \sigma}\right)^{C_2}\left(\frac{\rho_f}{\rho_g}\right)^{C_3+C_5}} \quad (1)$$

where: $C_1 = 0.0332$
$C_2 = -0.235$
$C_3 = -0.681$
$C_4 = 0.684$
$C_5 = 0.832$

The above equation was shown to correlate all of the data within ±20%.

Hall and Mudawar [21] have made a very important contribution to the literature by collecting and assessing 37,544 critical heat flux data points from the year 1949 onward, and covering more than 100 sources. The data are for water flow in uniformly heated vertical and horizontal tubes. It was reported that some of the data in the literature may be inaccurate because the errors that were introduced when the data were transported from the original sources. They identified a lack of CHF data for tubes smaller than 5 mm in diameter and for flow rates higher than 10,000 kg/m^{2}s. The authors also developed correlations for subcooled flow boiling CHF based both on inlet as well as outlet conditions. The correlation based on outlet conditions was thought to be most suited for tubes subjected to non uniform heat flux. It was claimed that these correlations were superior to the Look up Tables that have been proposed for use in *Thermal Hydraulic Analysis of Nuclear Reactors*. The correlation based on inlet conditions was similar to eq. (1) proposed earlier, but the five constants had the values:

$C_1 = 0.0722$
$C_2 = -0.312$
$C_3 = -0.644$
$C_4 = 0.900$
$C_5 = 0.724$

and correlated 4,860 data points with an r.m.s. error of 14.3%. The correlations based on outlet conditions was

$$\frac{q_{CHF}}{G_{hfg}} = C_1\left(\frac{G^2 D}{\rho_f \sigma}\right)^{C_2}\left(\frac{\rho_f}{\rho_g}\right)^{C_3}\left[1 - C_4\left(\frac{\rho_f}{\rho_g}\right)^{C_5} x_0\right] \quad (2)$$

where x_0 is thermodynamic equilibrium quality at exit. Equation (2) with constants having the same values as those for correlations based on inlet conditions, was found to correlate all of the data points with an r.m.s. value of 27.7%.

SUMMARY

A review of the studies performed at Purdue University during the last 15 years reveals that many fundamental contributions have been made in advancing the art and science of phase change heat transfer. These studies can be sub-divided into three general areas: material processing, microelectronic cooling, and phenomenological studies of subcooled critical heat flux. The Purdue Heat Transfer Group deserves our congratulations and gratitude for its vision, leadership, and seminal research in these areas.

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