

## A Study of Nucleate Boiling with Forced Convection in Microgravity

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### INTRODUCTION

The ultimate objective of basic studies of flow boiling in microgravity is to improve the understanding of the processes involved, as manifested by the ability to predict its behavior. This is not yet the case for boiling heat transfer even in earth gravity, despite the considerable research activity over the past 30 years. Hahne et al [1], for example, compared 7 different correlations with their own R12 forced convection boiling data, for both up and down flow, with distinctly different results.

The elements that constitute the nucleate boiling process – nucleation, growth, motion, and collapse of the vapor bubbles (if the bulk liquid is subcooled) – are common to both pool and flow boiling. It is well known that the imposition of bulk liquid motion affects the vapor bubble behavior relative to pool boiling, but does not appear to significantly influence the heat transfer. Indeed, it has been recommended in the past that empirical correlations or experimental data of pool boiling be used for design purposes with forced convection nucleate boiling [2, 3]. It is anticipated that such will most certainly not be possible for boiling in microgravity, based on observations made with pool boiling in microgravity, to be described below. In earth gravity buoyancy will act to remove the vapor bubbles from the vicinity of the heater surface regardless of how much the imposed bulk velocity is reduced, depending, of course, on the geometry of the system. The major so-called forces governing the motion of the bubbles are buoyancy, liquid momentum and viscosity. With sufficiently high flow Reynold's Numbers, it can be intuited that the latter two forces will outweigh the first, and the process will be the same whether at earth gravity or microgravity. However, as the Reynold's Number is reduced the magnitude of the liquid momentum and viscous forces are correspondingly reduced, and in microgravity buoyancy cannot take over as a "back-up" mechanism for vapor removal, leaving only the reduced levels of liquid momentum and viscous forces. Vapor bubbles have been observed to dramatically increase in size in pool boiling in microgravity [4], and the heat flux at which dryout took place was reduced considerably below what is generally termed the critical heat flux (CHF) in earth gravity, depending on the bulk liquid subcooling. However, at heat flux levels below dryout, the

nucleate pool boiling process was enhanced considerably over that in earth gravity [4,5], in spite of the large vapor bubbles formed in microgravity and perhaps as a consequence. These large vapor bubbles tended to remain in the vicinity of the heater surface, and the enhanced heat transfer appeared to be associated with the presence of what variously has been referred to as a liquid microlayer between the bubble and the heater surface. This layer serves as a boundary across which evaporation takes place, as well as a mechanism for the efficient removal of vapor bubbles from the heater surface, due to vapor pressure differences arising from surface tension.

Effects generally neglected at normal earth gravity, such as surface tension, both at the solid-liquid-vapor contact line and at the liquid-vapor surface associated with the interface temperature variation, become of consequence at microgravity conditions. The net quantitative effect of these on the vapor bubble behavior is unknown, at present, as are the related effects on the heat transfer, and provides one of the motivations for the study of the flow boiling process in microgravity.

The enhancement of the boiling process with low velocities in earth gravity for those orientations producing the formation of a liquid microlayer described above, accompanied by "sliding" vapor bubbles, has been demonstrated. The enhancement was presented as a function of orientation and subcooling in [6, 7], and as an additional function of heated length in [8,9], while a criterion for the heat transfer for mixed natural/forced convection nucleate boiling was given in [10].

A major unknown in the prediction and application of flow boiling heat transfer in microgravity is the upper limit of the heat flux for the onset of dryout (or critical heat flux – CHF), for given conditions of fluid-heater surfaces, including geometry, system pressure and bulk liquid subcooling. As stated above, it is clearly understood that the behavior in microgravity will be no different than on earth with sufficiently high flow velocities, and would require no space experimentation. However, the boundary at which this takes place is still an unknown. Furthermore, considering the high cost of pumping power in space, in terms of the availability of power, it can be anticipated that considerable effort will be expended in optimization

of the net energy requirements. This requires a sound understanding of the fundamental processes associated with the CHF.

Some results of CHF measurements were presented in [6] for low velocity flow boiling at various orientations in earth gravity as a function of flow velocity and bulk liquid subcooling. Preliminary measurements of bubble residence times on a flat heater surface at various orientations were given in [8] which showed promise as a parameter to be used in modeling the CHF, both in earth gravity and in microgravity. The objective of the work here is to draw attention to and show results of current modeling efforts for the CHF, with low velocities in earth gravity at different orientations and subcoolings.

### THE CRITICAL HEAT FLUX

Many geometrical possibilities for a heater surface exist in flowing boiling, with boiling on the inner and outer surfaces of tubes perhaps being the most common. If the vapor bubble residence time on and departure size from the heater surface bear a relationship to the CHF, as results to be given below indicate, it is important that visualization of and access to vapor bubble growth be conveniently available for research purposes. In addition, it is desirable to reduce the number of variables as much as possible in a fundamental study. These considerations dictated the use of a flat heater surface, as seen in the schematic of the test section in Figure 1.

The flat heater surface is rectangular in shape, 1.91 cm by 3.81 cm (0.75 x 1.5 inches), consisting either of a 400 Angstrom thick semi-transparent gold film sputtered on a quartz substrate which serves simultaneously as a heater and a resistance thermometer, or a copper substrate of the same size. The heater substrate is a disc which can be rotated so that the heated length in the flow direction can be changed from 1.91 to 3.81 cm (0.75 to 1.5 inches). The fluid is R-113, and the velocities can be varied between 0.5 cm/s and 60 cm/s. Details of both the experimental apparatus and model concepts to be outlined below are given in [11, 12].

For a sufficiently low velocity the CHF can be modeled reasonably well at various orientations by the correlation for pool boiling [13] corrected for the influence of bulk liquid subcooling [14], indicated by  $q_{co}$  in Figure 2, multiplied by the square root of  $\theta$  over the interval 90 to 270 deg. This arises from equating buoyancy and drag forces in the inverted positions where the vapor bubbles are held against the heater surface as they slide [15]. The angle  $\theta = 0$  applies to the horizontal upward facing orientation and  $\theta = 90$  to the vertical orientation with upflow.

A distortion of the measurements occurs to the right in Figure 2 as the flow velocity increases. In modeling this effect at different levels of subcooling it appeared appropriate to estimate the volumetric rate of vapor generation, using measurements of bubble frequency (or residence time), void fraction and average bubble boundary layer thickness. These were determined with the use of a platinum hot wire probe 0.025 mm in diameter by 1.3 mm long, applying a constant current to distinguish between contact with liquid or vapor. Two-dimensional spatial variations are obtained with a special mechanism to resolve displacements in increments of 0.025 mm. Figure 3 shows typical void fractions over a heater surface. From a number of such measurements it was determined that the fraction of the surface heat transfer resulting in evaporation varies inversely with the subcooling correction factor of [14] for the CHF.

The measured inverse bubble residence time  $1/\tau_{res}$  is normalized relative to that predicted for an infinite horizontal flat plate at the CHF [16], and is correlated well with the CHF normalized relative to that for pool boiling, for various orientation angles and subcooling levels, as seen in Figure 4. This correspondence is then combined with a normalizing factor for the energy flux leaving the heater surface at the CHF and the computed bubble radius at departure, determined from the balance between the outward velocity of the interface due to evaporation and the buoyancy induced velocity of the center of mass of the bubble. The product of the CHF and the corresponding residence time was determined to be a constant for all orientations at a given bulk flow velocity and liquid subcooling, and must be determined empirically for each velocity and subcooling at present.

It then becomes possible to predict the CHF for the different orientations, velocities, and subcoolings. These are shown in Figures 5-7 and compared with the normalized measurements for velocities ranging from 18 cm/s to 55 cm/s, over orientations  $\theta = 0$  to  $\theta = 360$ .

A direct comparison of the experimental data with the model predictions is given in Figure 8 for  $\theta = 0$  to  $\theta = 360$ , subcoolings from 2.8 to 22.2°K, and bulk velocities from 4 cm/s to 55 cm/s.

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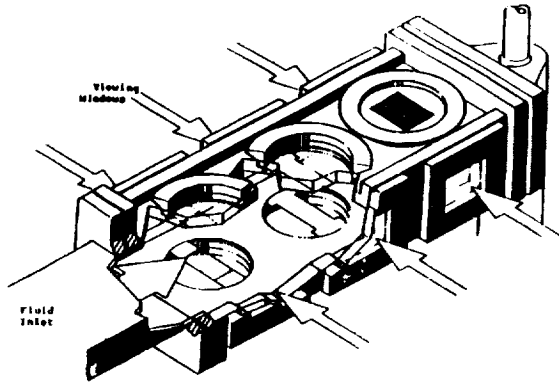


Figure 1. Test section. Flow area = 10.80 cm (4.25 in.) wide x 0.318, 1.27 or 2.54 cm (0.125, 0.50 or 1.0 in.) high.

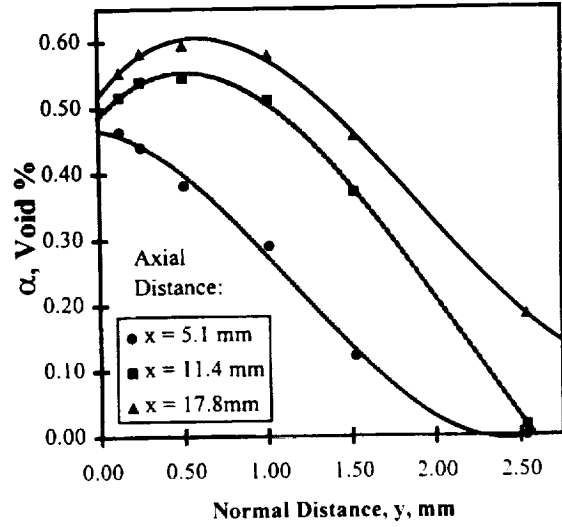


Figure 3. Void fraction profiles over the metal heater surface for  $\theta = 150$ ;  $U_{bulk} = 0.04 \text{ m s}^{-1}$ ;  $Re = 5400$ ;  $T_{in} = 322 \text{ K}$ ; 11.1 K subcooling;  $q_w'' = 175 \text{ kW m}^{-2}$ . R-113.

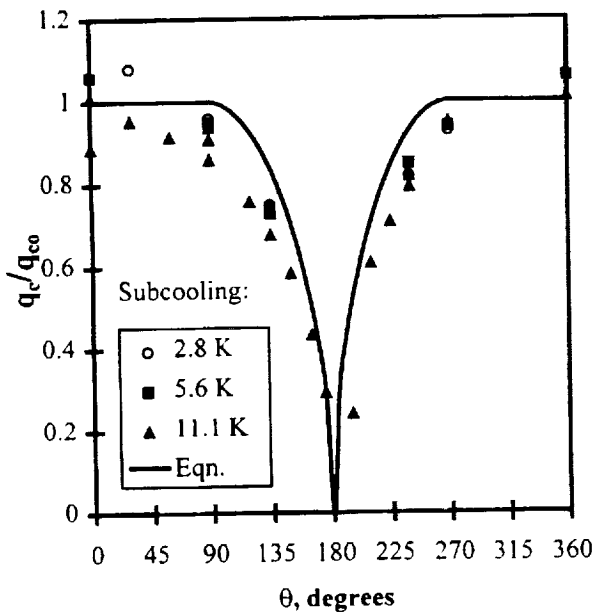


Figure 2. Measured CHF as a function of heater surface orientation compared with the CHF model: test conditions:  $U_{bulk} = 0.04 \text{ m s}^{-1}$ ;  $Re = 2700$ ;  $T_{in} = 322 \text{ K}$ . R-113.

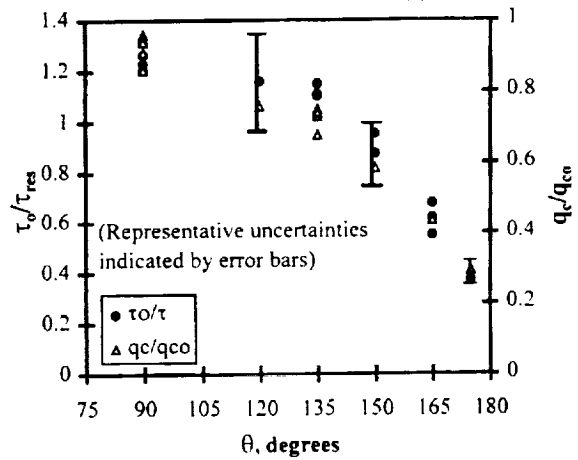


Figure 4. Reciprocal of measured bubble residence time as a function of heater surface orientation compared with corresponding CHF: test conditions:  $U_{bulk} = 0.04 \text{ m s}^{-1}$ ;  $Re = 5400$ ;  $T_{in} = 332 \text{ K}$ ; test section height = 25.4 mm. R-113.

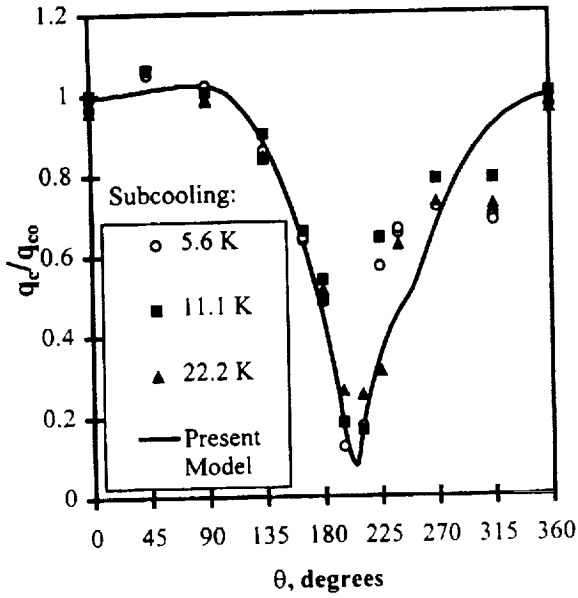


Figure 5. Comparison of measured CHF with present model prediction as a function of orientation: test conditions:  $U_{bulk} = 0.18 \text{ m s}^{-1}$ ;  $Re = 3400$ ; test section height = 3.2 mm;  $T_{in} = 322 \text{ K}$ . R-113.

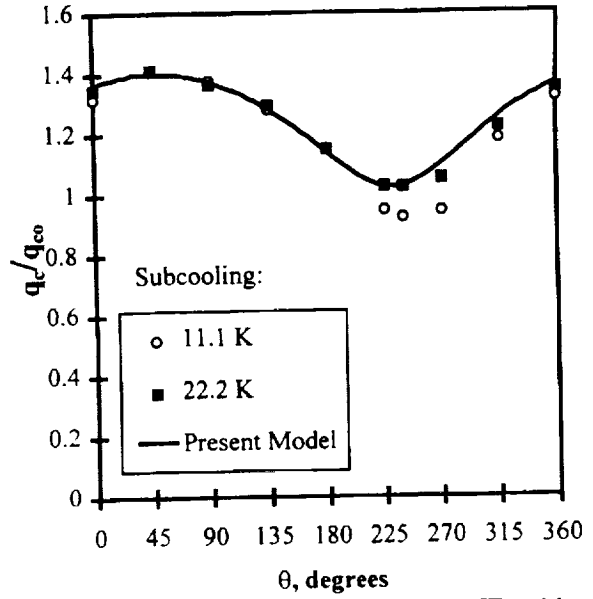


Figure 7. Comparison of measured CHF with present model prediction as a function of orientation: test conditions:  $U_{bulk} = 0.55 \text{ m s}^{-1}$ ;  $Re = 10500$ ; test section height = 3.2 mm;  $T_{in} = 322 \text{ K}$ . R-113.

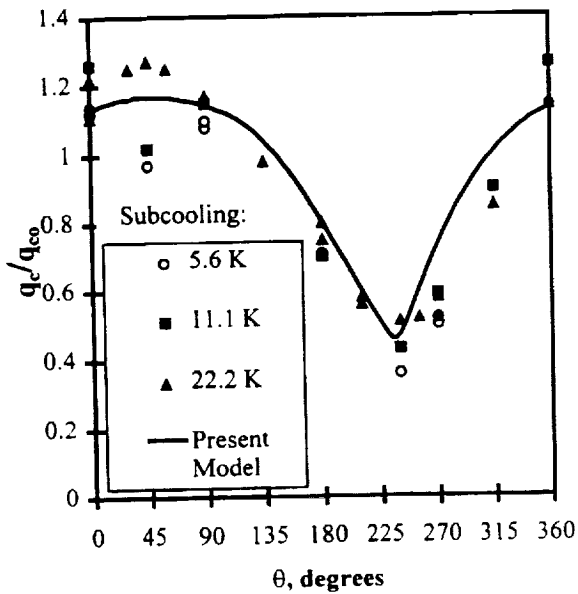


Figure 6. Comparison of measured CHF with present model prediction as a function of orientation: test conditions:  $U_{bulk} = 0.32 \text{ m s}^{-1}$ ;  $Re = 6300$ ; test section height = 3.2 mm;  $T_{in} = 322 \text{ K}$ . R-113.

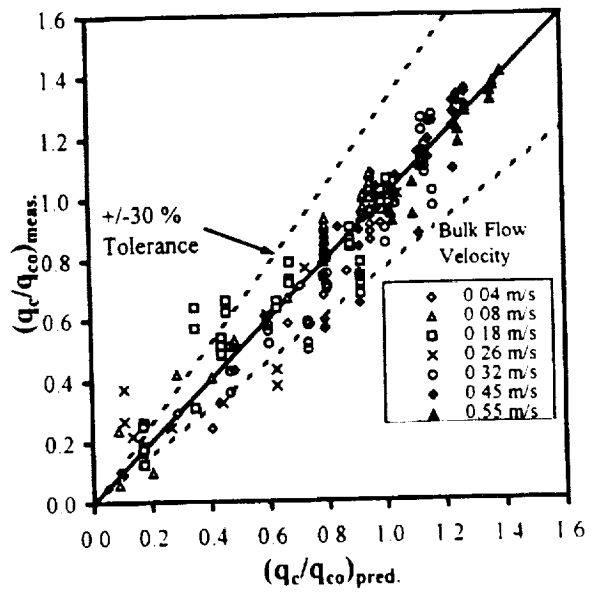


Figure 8. Comparison of measured CHF with present model predictions for bulk flow velocities ranging from  $0.04$  to  $0.55 \text{ m s}^{-1}$  and subcoolings ranging from  $2.8$  to  $22.2 \text{ K}$ . R-113.