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CRITICAL HEAT FLUX OF SATURATED CONVECTIVE BOILING ON UNIFORMLY HEATED PLATES IN A PARALLEL FLOW

Y. KATTO and C. KURATA
Department of Mechanical Engineering, University of Tokyo, Hongo, Bunkyo-ku, Tokyo, Japan

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Abstract—Employing saturated water and R-113 at atmospheric pressure, experiments are made for critical heat flux (CHF) on a uniformly heated plate of 10, 15 and 20 mm in length submerged parallel to a uniform liquid flow with velocity of 1.5–10 m/s, and the data of CHF obtained are successfully correlated by a generalized equation. In addition, it is shown that existing data of CHF for acetone, toluene, mono-isopropylbiphenyl and water flowing through internally heated annular channels of very small l/d_{he}, where l is the axial length and d_{he} the heated equivalent diameter, agree well with the above-mentioned correlation.

1. INTRODUCTION

Critical heat flux (CHF) in forced convection boiling ordinarily refers to internal flow in channels, and very few studies have been made on CHF of external flow. However, information about CHF for simple but fundamental boiling systems utilizing external flow are of importance not only for industrial applications (augmentation of heat removal from various heated surfaces with boiling, for example) but also for the potential insight into the basic mechanisms of forced convection boiling.

Among the boiling systems utilizing external flow, one of the simplest is the boiling of a thin liquid film flowing over a uniformly heated surface (exactly, a downward-facing horizontal surface without reentry of splashed liquid to the heated surface). For this type of boiling, Monde & Katto (1978), and Katto & Shimizu (1979) measured CHF for disk heaters cooled by a small round jet of saturated (and subcooled) liquid yielding generalized correlations of CHF data, while Lienhard & Eichorn (1979), and Lienhard & Hasan (1979), analyzed these data theoretically. On the other hand, Katto & Ishii (1978) determined CHF experimentally for forced flow of a plane jet of saturated liquid over a rectangular heater, and presented a generalized correlation for CHF data obtained.

Meanwhile, for forced convection boiling on a heated surface submerged in liquids of sufficiently large volume, Andrews & Mohan Rao (1974) made experiments on CHF for thin horizontal ribbons in submerged vertical liquid jets. The experiments of CHF on a cylinder normal to the liquid flow were also made by McKee & Bell (1969) and others, and recently Lienhard & Eichorn (1976) carried out a theoretical analysis along with an experimental study with saturated liquids, yielding generalized correlations of CHF data in two characteristic regimes.

On the other hand, it is noted that there is a much simpler submerged system, that is, boiling on a uniformly heated plate in a parallel flow, and to the authors' knowledge, experiments have scarcely been made for this system. In the present study, therefore, experiments are made for CHF of this boiling system with saturated water and R-113 at atmospheric pressure.

2. EXPERIMENTAL APPARATUS

Figure 1 shows the test vessel. A copper block is heated by eleven plate heaters inserted into grooves on the l.h.s. Boiling takes place on a rectangular flat surface at the right end of the copper block with a width of 10 mm and three lengths $l = 10$, 15 and 20 mm. Three Chromel–Alumel thermocouples embedded along the central axis of the copper block are used to

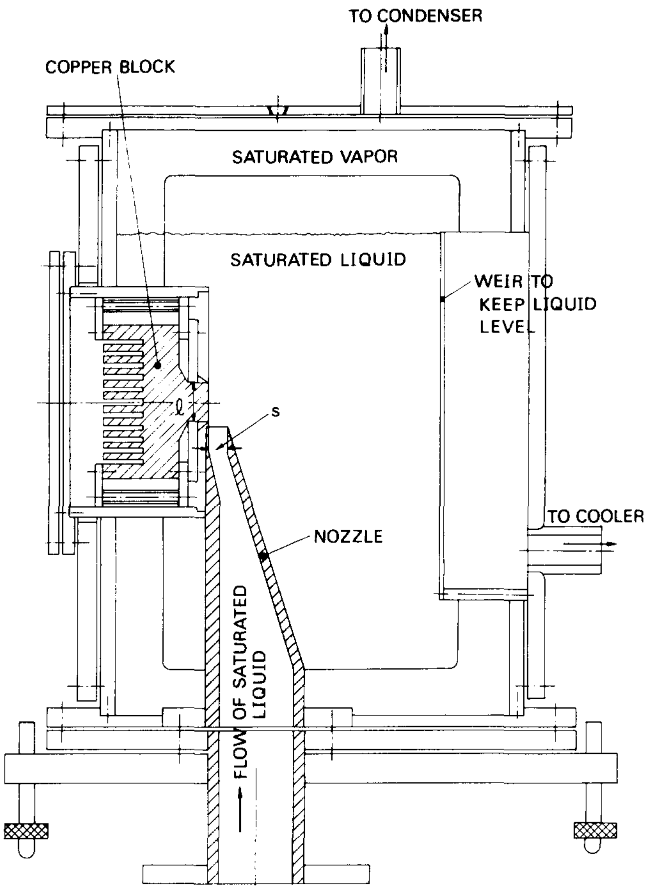

Figure 1. Test vessel.

measure the heat flux across the heated surface as well as to detect the occurrence of CHF indicated by a sharp rise in temperature.

In the test vessel, a nearly constant volume of liquid is maintained at atmospheric pressure by means of a weir on the r.h.s. An auxiliary electric heater, omitted from figure 1, is used to keep the liquid at saturation temperature. Meanwhile, practically saturated liquid with a subcooling of less than 3°C, that is fed through a volute pump, a flow meter and a preheater, flows out of the exit of a rectangular nozzle shown in figure 1 (with a width of 15 mm, and two heights $s = 10$ mm mainly and 5 mm auxiliary) to give a forced convection flow of prescribed velocity along the heated surface. According to the experimental results mentioned in the next chapter, it is found that the difference of CHF is hardly perceived for the change of s from 10 to 5 mm, so that it may be assumed that the experimental condition employed in this study is a very close approximation to that of the heated flat surface parallel to a uniform velocity flow of liquid. Finally, the cooler noted in figure 1 is necessary to remove at least the energy supplied to the system through the dissipation of pump power.

3. EXPERIMENTAL RESULTS OF CHF

3.1 *CHF and flow pattern*

Experiments were made in the range of liquid velocity $u = 1.5$–10 m/s at the nozzle exit for both water and R-113. According to still pictures taken of the two-phase flow along the heated surface, the flow pattern is bubbly when the heat flux q is low, but as q is increased, coalescence of vapor takes place near the heated surface presenting the appearance shown in figure 2(a).

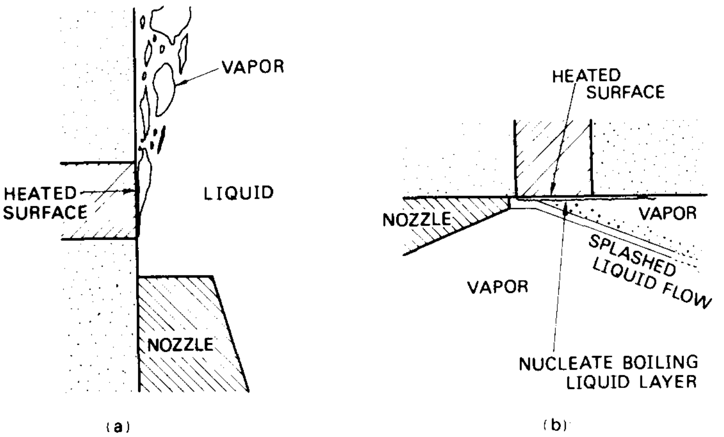

Figure 2. Flow pattern near CHF condition: (a) forced flow of liquid with uniform velocity, (b) forced flow of liquid film.

3.2 *Correlation of experimental data*

All the data of CHF obtained in this study as listed in tables 1 and 2 are plotted in the dimensionless form of q_{co}/GH_{fg} against $\sigma\rho_l/G^2l$ to give figure 3, where q_{co} is the critical heat flux for saturated liquid flow, G the mass velocity of liquid flowing out of the nozzle, H_{fg} the latent heat of evaporation, σ the surface tension, ρ_l the density of liquid, and l the length of heated surface in the direction of flow. In figure 3, data for water and R-113 appear to fall along lines having nearly the same gradient. If a distinction is attempted between the data of $s = 10$ mm and those of $s = 5$ mm in figure 3, the former group of data seem to have a slightly greater gradient than the latter. However, the difference in gradient, if any, is very small. There is not really sufficient data to analyze this effect, so the difference in gradient is neglected in this paper. Then, the situation in figure 3 is quite similar to those encountered in the previous studies (Monde & Katto 1978, Katto & Ishii 1978), suggesting the possibility of unifying the data of water and R-113 by taking into account the effect of vapor/liquid density ratio ρ_v/ρ_l; namely, if $(q_{co}/GH_{fg})/(\rho_v/\rho_l)^{0.559}$ is used instead of q_{co}/GH_{fg} for the ordinate of figure 3, two separate groups of data come together to give the result of figure 4, through which a generalized correlation of CHF in the experimental range of $\sigma\rho_l/G^2l = 10^{-5}$ to 3×10^{-3} is given as follows:

$$\frac{q_{co}}{GH_{fg}} = 0.186\left(\frac{\rho_v}{\rho_l}\right)^{0.559}\left(\frac{\sigma\rho_l}{G^2l}\right)^{0.264}. \qquad [1]$$

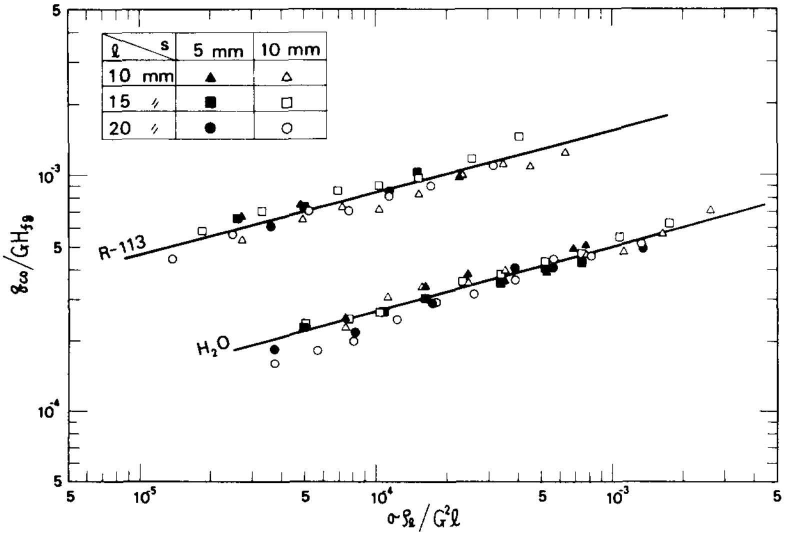

Figure 3. Experimental data of CHF.

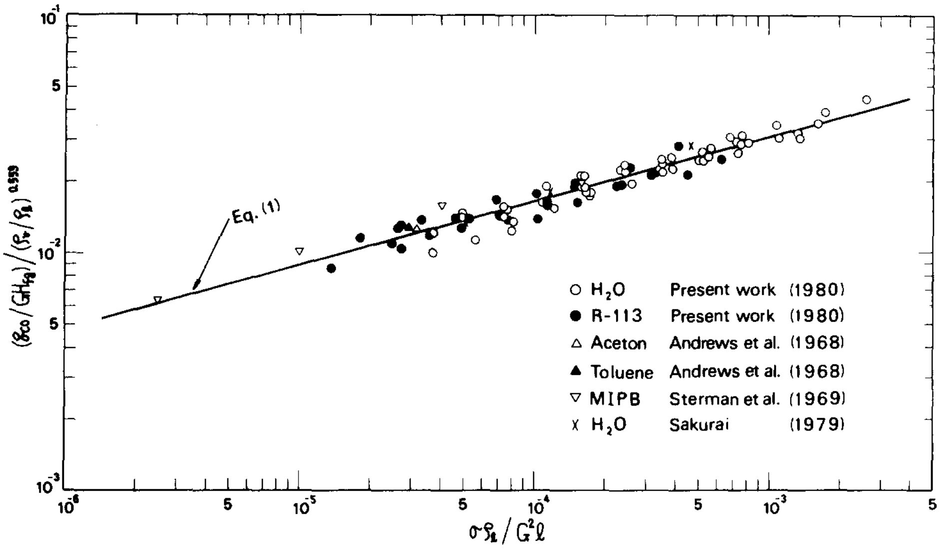

Figure 4. Generalized correlation of CHF data.

Table 1. Experimental data of CHF of water

l = 10 mm				l = 15 mm				l = 20 mm			
s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm	
u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2
2.83	3.08	1.53	2.36	2.35	2.15	1.53	2.06	1.51	1.59	1.52	1.69
3.00	3.20	1.95	2.37	2.79	2.42	1.95	2.34	2.35	2.06	1.95	1.97
3.42	2.88	2.35	2.47	3.45	2.67	2.35	2.36	2.83	2.47	2.35	2.24
4.18	3.52	2.84	2.84	5.02	3.29	2.83	2.59	4.18	2.62	2.81	2.19
5.02	4.12	3.42	2.90	6.13	3.50	3.43	2.83	6.14	2.91	3.43	2.33
6.14	4.52	4.19	3.59	9.09	4.47	4.19	3.28	9.09	3.84	4.19	2.59
9.09	4.95	5.02	3.86			5.02	3.30			5.00	2.69
		6.18	4.59			6.17	3.51			6.18	2.66
		7.37	4.90			7.37	3.97			7.37	2.91
		9.11	4.49			9.10	4.64			9.11	3.14

Table 2. Experimental data of CHF of R-113

l = 10 mm				l = 15 mm				l = 20 mm			
s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm		s = 5 mm		s = 10 mm	
u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2	u m/s	q_{co} MW/m^2
2.10	0.453	1.26	0.349	2.10	0.472	1.28	0.412	2.10	0.394	1.26	0.306
4.54	0.753	1.49	0.360	3.72	0.599	1.62	0.421			1.70	0.338
6.06	0.897	1.70	0.418	5.00	0.731	2.10	0.454			2.10	0.378
		2.10	0.451			2.55	0.510			2.55	0.399
		2.55	0.456			3.10	0.591			3.10	0.491
		3.13	0.493			3.70	0.606			3.71	0.502
		3.71	0.600			4.48	0.698			4.48	0.554
		4.48	0.647			6.02	0.782			6.02	0.590
		6.02	0.713								

4. CHF FOR ANNULI OF VERY SMALL l/d_{he}

In the case of an annular channel with a uniformly heated rod, if l/d_{he} (where l is the axial length of heated rod and d_{he} the heated equivalent diameter) is extremely small, the condition of convective boiling is closer to that dealt with in this paper than that in annular channels, as pointed out by Katto (1980). From this point of view, therefore, existing data of CHF obtained for internally heated annuli of $l/d_{he} < 10$ (see Katto 1980) will be utilized below to check the applicability of [1]

Table 3. Values of ρ_v/ρ_l for CHF data

Source	Fluid	Pressure	$\rho_v/\rho_l \times 10^2$	l/d_{he}
Andrews et al.(1968)	Acetone	1 atm	0.291	1.15
Andrews et al.(1968)	Toluene	1 atm	0.385	1.15
Sterman et al.(1969)	MIPB*	2.0 bar	1.03	7.05
Sakurai (1979)	Water	1.43bar	0.0868	0.0582
Present work	Water	1 atm	0.0624	-----
Present work	R-113	1 atm	0.490	-----

* Monoisopropylbiphenyl.

For annuli of very small l/d_{he} with internal heating, experimental data of CHF have been given as a function of inlet subcooling enthalpy ΔH_i by Andrews *et al.* (1968) for acetone and toluene, and by Sterman *et al.* (1969) for monoisopropylbiphenyl (MIPB), so that the critical heat flux q_{co} for saturated inlet conditions can be estimated by the extrapolation as $\Delta H_i \to 0$. Sakurai (1979) has measured CHF for saturated water at the pressure of 1.43 bar. Table 3 lists the values of ρ_v/ρ_l along with l/d_{he} for the experiments on annuli mentioned above, where physical properties of MIPB are estimated from Core & Sato (1958) and Gambill (1957, 1958, 1959). It also lists the values of ρ_v/ρ_l for the present work. The data for CHF in annuli are shown in figure 4, revealing that it is within the scatter of the data from the present work, though there is a trend of appearing slightly higher than the prediction of [1].

5. DISCUSSIONS

(i) The design of the flow channel near the nozzel exit adopted in figure 1 to give uniform velocity at the nozzle exit, may raise question concerning the velocity component perpendicular to the heated surface. However, according to a simple analysis, the difference in dynamic pressure due to this cause is, at most, a few per cent. In addition, the CHF data for annuli, which are free from the above problem, agree fairly well with [1] as seen in figure 4, suggesting that the design of the flow channel near the nozzle exit does not seriously affect the results.

(ii) The generalized correlation of CHF determined experimentally by Katto & Ishii (1978) for the flow of thin plane jet of saturated liquid over rectangular heated surface is written as:

$$\frac{q_{co}}{GH_{fg}} = 0.0164 \left(\frac{\rho_v}{\rho_l}\right)^{0.133} \left(\frac{\sigma\rho_l}{G^2 l}\right)^{1/3}. \qquad [2]$$

Comparing this equation with [1] reveals that the exponent of the reciprocal Weber number $\sigma\rho_l/G^2 l$ is nearly the same between the two equations with only a difference of 20 per cent, whereas the difference in exponent of density ratio ρ_v/ρ_l is considerable. In other words, the effect of ρ_v/ρ_l on CHF is much greater in the case of a submerged heated surface than in the case of thin liquid film flow. Meanwhile, if the flow pattern in the former case such as shown in figure 2(a) is compared with that in the latter case such as shown in figure 2(b), which is reproduced from the paper of Katto & Ishii (1978), it may suggest that the escape of generated vapor from the heated surface is much more difficult in the former case due to the interference of a greater mass of liquid. Thus it may be natural to suppose that the high value of exponent of ρ_v/ρ_l in [1] relates mainly to the flow pattern near the heated surface submerged in liquid.

(iii) For CHF on a uniformly heated cylindrical surface in a cross flow, Lienhard & Eichorn (1976) gave generalized correlation equations valid in two regimes, namely the low-velocity and high velocity regimes. According to their observations, both regimes are characterized by two-dimensional flow of escaping vapor. If the present nomenclature is employed, their equations can be written as follows:

For the low-velocity regime,

$$\frac{q_{co}}{GH_{fg}}=\frac{1}{\pi}\left\{\frac{\rho_v}{\rho_l}+4^{1/3}\left(\frac{\rho_v}{\rho_l}\right)^{2/3}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}\right\} \qquad [3]$$

and for the high-velocity regime,

$$\frac{q_{co}}{GH_{fg}}=\frac{1}{\pi}\left\{\frac{1}{169}\left(\frac{\rho_v}{\rho_l}\right)^{1/4}+\frac{1}{18.2}\left(\frac{\rho_v}{\rho_l}\right)^{1/6}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}\right\} \qquad [4]$$

where D is the diameter of cylinder.

Both [3] and [4] are, however, too complicated in form for the purpose of comparison with [1] and [2]. Therefore, apart from the theoretical analysis of CHF made by Lienhard and Eichorn, [3] and [4] will be replaced by the following empirical equations only for convenience. According to figure 8 in the paper of Lienhard & Eichorn (1976), [3] may be replaced, keeping the same accuracy, by

$$\frac{q_{co}}{GH_{fg}}=0.530\left(\frac{\rho_v}{\rho_l}\right)^{2/3}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3} \qquad [3']$$

and from figure 7 in the same paper, suggesting a trend that $q_{co}/GH_{fg}\rightarrow 0$ as $\sigma\rho_l/G^2l\rightarrow 0$, it is found that [4] may be replaced by

$$\frac{q_{co}}{GH_{fg}}=0.0202\left(\frac{\rho_v}{\rho_l}\right)^{1/6}\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}. \qquad [4']$$

Equations [3′] and [4′] are very close to the respective equations obtained by ignoring the first terms on the r.h.s. of [3] and [4]. In addition, taking into account the results of [3′] and [4′] as well we the criterion assumed by Lienhard and Eichorn for the transition between the low-velocity and high-velocity regimes, it is concluded that [3′] applies to the low-velocity range designated by

$$\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}>\frac{1}{0.530\pi}\left\{\left(\frac{\rho_v}{\rho_l}\right)^{1/3}+0.275\left(\frac{\rho_v}{\rho_l}\right)^{-1/6}\right\} \qquad [5]$$

and [4′] applies to the high-velocity range designated by

$$\left(\frac{\sigma\rho_l}{G^2D}\right)^{1/3}<\frac{1}{0.0202\pi}\left\{\left(\frac{\rho_v}{\rho_l}\right)^{5/6}+0.275\left(\frac{\rho_v}{\rho_l}\right)^{1/3}\right\}. \qquad [6]$$

Now, [3′] and [4′] can be compared with [1] and [2] to reveal the interesting fact that [3′] for CHF on cylinders in the low-velocity regime bears similarities to [1], whereas [4′] for CHF on cylinders in the high-velocity regime is very nearly related to [2]. Thus, in the case of low-velocity, the vapor removel process from the back surface as well as from the front surface of cylinder may be supposed to remain the type of froth flow shown in figure 2(a). For reference, the presumption of flow pattern in this case is shown in figure 5(a). On the other hand, in the case of high-velocity, 2-dimensional vapor sheet develops behind the cylinder as shown in figure 5(b), so that the back surface of cylinder is covered with a nucleate boiling liquid layer in the similar way as in figure 2(b), and the onset of CHF condition is assumed to

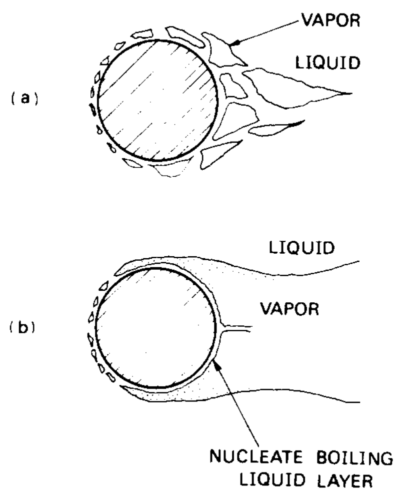

Figure 5. Presumption of flow pattern near CHF condition for nucleate boiling on a heated cylindrical surface in a cross flow: (a) low-velocity regime, (b) high-velocity regime.

occur first at the rear stagnation line. It has been observed by Katto & Ishii (1978) that in case of figure 2(b), the CHF condition occurs through the breakdown of the nucleate boiling liquid layer at the rear edge of heated surface.

5. CONCLUSIONS

Equation [1] has been obtained as a generalized correlation of CHF of saturated convective boiling on heated surfaces submerged parallel to a liquid flow of uniform velocity. Taking into account the data for internally heated annular channels of very small l/d_{he} as well as the data obtained in this study, the experimental range of ρ_v/ρ_l and $\sigma\rho_l/G^2l$ for which [1] is valid is as follows: $\rho_v/\rho_l = 0.000624\text{–}0.0103$, and $\sigma\rho_l/G^2l = 2\times10^{-6}\text{–}3\times10^{-3}$.

Then, on the basis of two typical flow patterns shown in figure 2, presumative discussions are made on the relation between [1] for heated plates in a parallel flow, [2] for heated plates covered with a liquid film flow, and [3′] and [4′] for heated cylinders in a cross flow.

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