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Critical heat flux as a mass flux dependent local or global phenomenon: Theoretical analysis and experimental confirmation

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Abstract

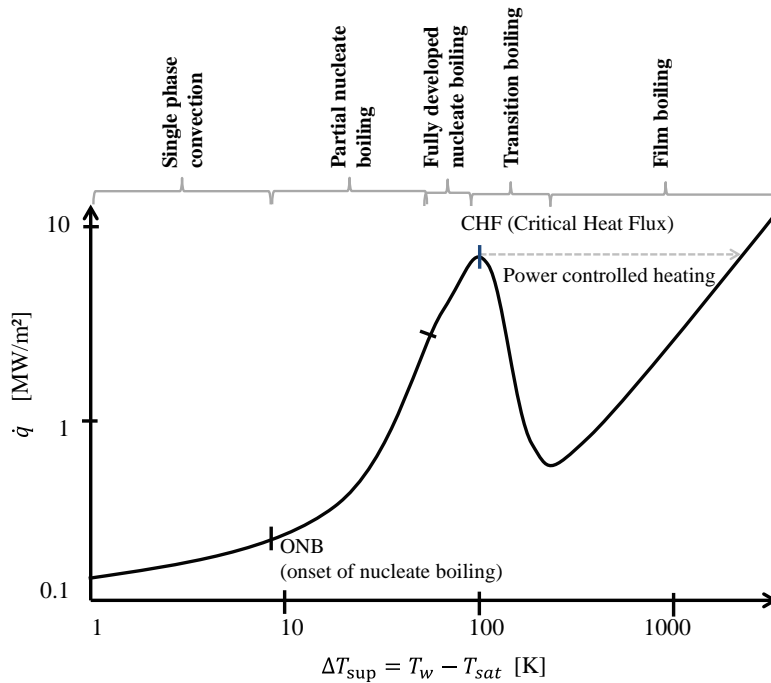
In this article, we report on a theoretical analysis and experimental investigations on critical heat flux (CHF) in subcooled flow boiling. Commonly, CHF is considered as a local phenomenon. A validated CHF- concept recently developed in our group indicated that CHF may be initiated in two different ways, that is, locally and globally. We designed and conducted an experiment to verify this hypothesis. The experimental results agree well with the expectations from our CHF- modelling and confirm the two mechanisms. Following that, we continued to clarify the role of different parameters, such as channel orientation, channel length and hydraulic diameter. The new concept of CHF is useful to explain and predict CHF at conditions of low pressure and low fluid velocity.

Keywords: boiling, critical heat flux, initiation mechanisms

1. Introduction

Subcooled flow boiling is one of the most efficient ways to transfer heat, as it combines the uptake of latent heat by bubbles, convective heat transfer via bubble motion and an effective mixing in the thermal boundary layers. However, when the heat flux reaches a critical value, the so-called Critical Heat Flux (CHF), nucleate boiling turns into film boiling. There, parts of the heater surface become irreversibly covered by a thin vapour blanket, which lowers the heat transfer drastically [1] (Figure 1). In power-controlled systems this may jeopardizes the safety as it can potentially lead to a meltdown of the heater. Because of this, the transition from nucleate to film boiling at CHF is still a topic of intense scientific investigation. Though many experimental studies and mechanistic models do exist, a comprehensive understanding of CHF has not yet been achieved. One difficulty comes from the fact, that an optical observation of critical heat flux on metallic heater surfaces is problematic, as the heavy pre-CHF boiling makes the heat transfer fluid opaque and further harsh pressure and temperature conditions hamper the application of measurement techniques.

From many experimental studies it is known that a number of system parameters, such as channel orientation, channel length and hydraulic diameter have a different influence on the CHF under high and low mass flux in flow boiling. However, mechanistic models [2-5] do not consider such effects. In our opinion, the reason for that is that present mechanistic models do consider CHF as a local phenomenon. Our recent analyses, however, indicate that CHF may occur in two different ways, namely in a local and a global one, which should be dependent on the system's mass flux. Local occurrence is happening at low mass flux while at high mass flux CHF appears globally. In this article, we report on a theoretical and experimental study to clarify this issue. Before that, we will firstly give a brief introduction of common knowledge and recent findings in the field.



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37
38 *Figure 1: Exemplary boiling curve showing the transitions from single-phase convection to film boiling*

39 **1.1 Impacts of various parameters on CHF**

40 In 1963, Bergles [6] identified six main system parameters affecting the CHF: pressure (p), liquid subcooling
 41 (ΔT_{sub}), mass flux (G , in subcooled flow boiling), channel length (L), hydraulic diameter (D) of channel (in
 42 subcooled flow boiling) and channel orientation (ϕ). He reported that the pressure has a rather weak influence
 43 on CHF. That is, for forced convective boiling of water CHF increases around 17% when the pressure changes
 44 from 0.14 MPa to 0.6 MPa. Vandervort et al. [7] and Celata [8] claimed that the pressure has even no significant
 45 effect on CHF when it is below 2.5 MPa. Sakurai and Shiotsu [9] did experiments to investigate the impact of
 46 subcooling on CHF. They found a linear relationship between CHF and subcooling in the entire subcooling
 47 region up to saturation. They concluded that the CHF increases with an increase of subcooling in horizontal and
 48 vertical pool boiling. Gunther [10] found for flow boiling of water in a rectangular section that CHF also has a
 49 linear relationship with subcooling. Bergles found a similar effect but only at relatively high subcooling. Celata
 50 and Mariani [2] conducted experiments to study the impact of mass flux on CHF. They indicated that high mass
 51 flux could lead to higher CHF, which was widely agreed by others [11-13]. Bergles experimentally observed an
 52 impact of the pipe diameter (D) on CHF. He stated that CHF increases with decreasing D . However, this effect
 53 becomes less significant for hydraulic diameters above 5 mm. Nariai et al. [3], and Nariai and Inasaka [4]
 54 conducted experiments to investigate the effect of the channel length (L) and channel diameter (D) on CHF.
 55 They found that at high mass flux, CHF increases when both channel diameter and length decrease. However, at
 56 low mass flux, both effects become insignificant for sufficiently high values of L and D . They also found that
 57 CHF values are higher at high mass flux and at low mass flux with small L and D . Buoyancy and channel
 58 orientation also influence CHF. Pappell et al. [5] demonstrated in experiments with liquid nitrogen in a 12.5 mm
 59 diameter tube that buoyancy only plays a role for low mass flux. Celata and Mariani [2] found that the channel
 60 orientation plays a role only when mass flux is small. They proposed a criterion based on the comparison of the
 61 buoyancy to inertia forces basing on the modified Froude number

62

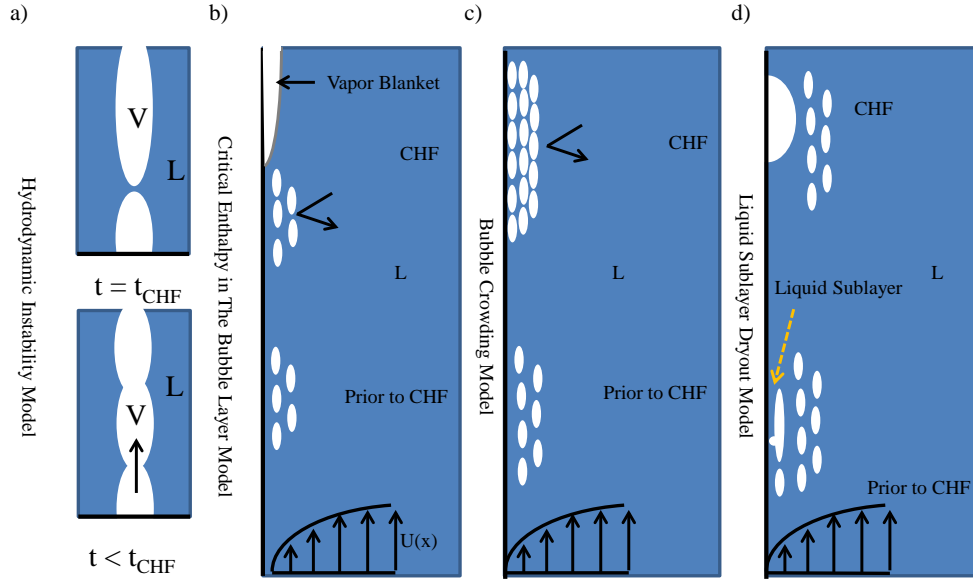
$$Fr = \frac{G \cos \phi}{\rho_l \left[g D \left(\frac{\rho_l - \rho_g}{\rho_l} \right) \right]^{1/2}}, \quad (1)$$

63

64 where G is the mass flux, ϕ is the orientation angle, ρ_l and ρ_g are the liquid and vapor density, g is the
 65 gravitational acceleration and D is tube diameter. They concluded that the effects due to orientation only appear
 66 when the modified Froude number is smaller than 5. While the dependency of CHF on the above parameters is
 67 well recognized, the mechanisms are still not fully included in the mechanistic models of CHF.

68 1.2 Mechanistic models and correlations

69 At present, the most popular mechanistic models of CHF can be classified into four categories [2, 13]: the
 70 *hydrodynamic instability model*, the *model of critical enthalpy in the bubble layer*, the *model of vapor removal*
 71 *limit and near-wall bubble crowding model*, and the *liquid sublayer dryout model* (Figure 2). Kutateladze [14]
 72 and other researchers [15, 16] hypothesized that CHF of saturated horizontal pool boiling is a purely
 73 hydrodynamic phenomenon and that it is triggered by a destruction of the stability of the two-phase flow
 74 occurring close to the heated surface (Figure 2a). In 1966, Ivey and Morris [17] further extended this model for
 75 subcooled boiling. The '*model of critical enthalpy in the bubble layer*' was proposed by Tong et al. [18]. They
 76 considered that a bubbly layer near the heater surface could trap the liquid in between. CHF is reached when this
 77 liquid layer attains a certain limiting enthalpy (Figure 2b). The '*bubble crowding model*' was proposed by Hebel
 78 et al. [19] who considered the limit of the turbulent interchange between the bubbly layer and the bulk of the
 79 liquid and inferred that crowding of the bubbles prevents the liquid access to the heater wall (Figure 2c).
 80 Weisman and Pei [20] further quantified this model by the assumption that CHF occurs when the volume
 81 fraction of vapor in this bubbly layer exceeds a critical volume fraction of 82%. This definition is based on their
 82 experimental observations. The '*liquid sublayer dryout model*' was proposed by Katto and Yokaya [21] and
 83 further developed by Haramura and Katto [22], Lee and Mudawar [23] and Celata et al [24]. This model
 84 assumes that a liquid film forms near the heater wall because of a Helmholtz instability (Figure 2d) and CHF is
 85 reached when the heater can provide the necessary latent heat to completely evaporate the liquid entering the
 86 film between the liquid sublayer and wall. All these models can to some degree achieve agreement with
 87 experimental data, but they contain quite a few empirical constants or empirical correlations [13]. As many of
 88 these model concepts and correlations are for subcooling boiling at high pressure and high velocity (HPHV) they
 89 typically produce larger disagreement at low pressure and low velocity conditions [12]. Moreover, the existing
 90 model concepts do mostly not contain the dependencies on the above-mentioned system parameters, such as
 91 channel orientations, channel length and hydraulic diameter on CHF at low and high mass flux.



92

93 *Figure 2: CHF mechanisms according to a) the hydrodynamic instability model, b) the model of critical*
 94 *enthalpy in the bubble layer, c) the bubble crowding model, d) the liquid sublayer dryout model. V stands for*
 95 *vapor, L stands for liquid and the turning arrows indicate partial blockage of liquid supply to the wall.*

96

2. CHF- Concept and Model

97 Recently, we developed a model of prior critical heat flux (CHF-) from the models of bubble dynamics at
 98 nucleate boiling [25]. It holds for pool boiling and forced convective boiling and incorporates a mutual effect
 99 model and a shear stress model. The model is capable to explain the initiation mechanisms of the boiling crisis
 100 and impacts of different variables. In the following section, the main idea of this CHF- concept will be briefly
 101 introduced.

102 The bubble growth at a small cavity in the heater wall is considered as a stable and repeatable process which
 103 consists of activation, growth, departure and reactivation with certain durations (Figure 3). Most important
 104 periods are the total growth period t_g and the waiting time t_w between bubble departure and new activation.

105 The total transferred heat Q_b during bubble growth consists of three parts: heat flowing from the wall into the
 106 bubble via evaporation ($Q_{b,w}$), heat flowing from the superheated liquid near the wall into the bubble ($Q_{b,s}$) and
 107 condensation heat transfer to the bulk liquid at the upper part of the bubble ($Q_{b,c}$). That is, the heat input to the
 108 bubble comes from two sources: the wall and the thermal boundary layer, though we cannot say for the moment,
 109 how the share is quantitatively. After bubble departure, a waiting time is required to regenerate the nucleus in the
 110 cavity and to recover the thermal layer. In this time, the quenching heat Q_q is delivered from the wall to the
 111 liquid. As during bubble growth the liquid in the vicinity of the bubble has gained the heat $Q_{b,c}$ and lost the
 112 heat $Q_{b,s}$, we may assume that $Q_q = Q_{b,s} - Q_{b,c}$. The heat balance can then be written as

113

$$Q_{b,w} + Q_q = Q_{b,w} + Q_{b,s} + Q_{b,c} = Q_b = \frac{4}{3}\pi r_d^3 \rho_g h_{fg}, \quad (2)$$

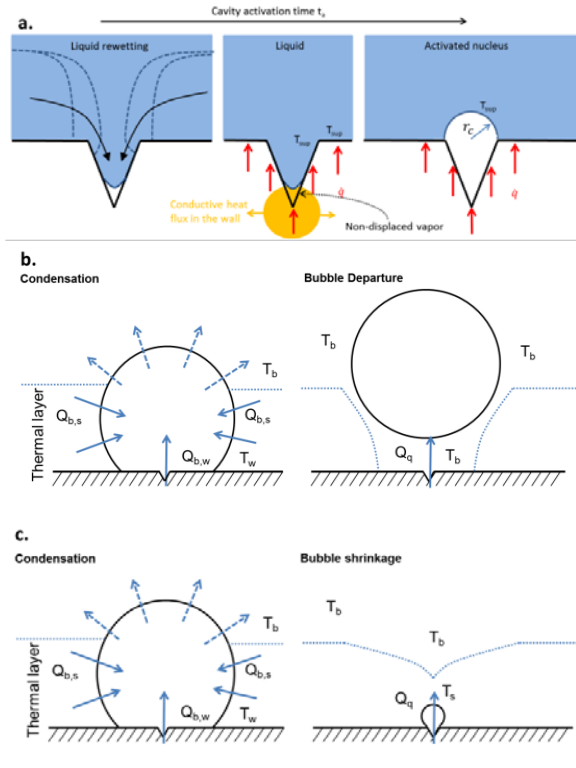
114

115 with h_{fg} being the latent heat of the fluid.

116 Then we define the projective area $A_b = \pi r_d^2$ of a fully developed bubble with departure radius r_d as the
 117 apparent heat transfer area for boiling heat transfer per single bubble. The heat flux in this area during bubble
 118 growth is then
 119

$$\dot{q} = \frac{Q_{b,w} + Q_q}{\pi r_d^2 (t_g + t_c + t_w)} = \frac{Q_b}{\pi r_d^2 (t_g + t_c + t_w)} = \frac{\left(\frac{4}{3}\pi r_d^3 \rho_g h_{fg}\right)}{\pi r_d^2 (t_g + t_c + t_w)} \quad (3)$$

120
 121 where t_c is the condensation time which is only for the case where the bubble will activate, grow, shrink and
 122 collapse on the cavity in high subcooling as shown in Figure 3 c). For saturation or low subcooling conditions
 123 we may set $t_c = 0$.
 124



125
 126
 127
 128 *Figure 3: a) Cavity activation and heat transport during and after bubble growth: b) bubble departure in the*
 129 *low subcooling case and c) bubble shrinkage in the high subcooling case in horizontal pool boiling [25].*

130 During rewetting the replenishing liquid partially displaces the vapor in the cavity [26, 27]. Before the bubble
 131 cycle closes it takes a time period t_a (cavity activation time) to form a new vapor nucleus with critical radius
 132 r_c on the mouth of cavity, which is the activation condition. The formation of this vapor nucleus requires
 133 superheating of liquid in the cavity to overcome the high pressure in the nucleus. In the upper part of the cavity
 134 liquid touches the wall and we may assume that heating up this liquid has the same time scale as above the wall
 135 outside the cavity. In the lower part of the cavity there is vapor in direct contact with the cavity wall, which
 136 gives a low heat transfer coefficient. Hence we have a microscopic “film boiling” situation on the bottom of the
 137 cavity, which leads to a stronger superheat of the wall there. Conductive heat transfer towards the liquid through
 138 the wall material is then the major heat removal mechanism. The higher the heating power the higher becomes
 139 the wall superheat at the bottom of the cavity. On the other hand, as the vapor in the lower part of the cavity
 140 presents a strong heat transfer resistance, the liquid in the center of the cavity is heated slower than in single

141 phase conditions and evaporation into the nucleus is delayed. This effect determines the activation time t_a . As
 142 the vapor pocket delays the heating of the liquid in the cavity, we can assume that the recovery of the thermal
 143 boundary layer (recovery time t_r) occurs faster than the reactivation of the cavity, that is $t_a > t_r$. With
 144 increasing heating power or heat flux this situation can change. The wall material at the bottom of the cavity will
 145 superheat so much, that reheating of liquid in the cavity is again synchronous with the build-up of the thermal
 146 boundary layer outside the cavity, that is, $t_a = t_r$. If the heat flux further increases, the activation time becomes
 147 even shorter, the vapor nucleus grows rapidly into the yet not established thermal boundary layer and transfers
 148 additional heat to the boundary layer, then $t_a < t_r$. The larger one of t_r and t_a decides the total waiting time,
 149 that is, $t_w = \max(t_a, t_r)$. For $t_a < t_r$ the situation is unstable, as increasing vapor content in the cavity will
 150 further increase cavity wall superheat and further shorten t_a . Eventually the temperature of the cavity wall and
 151 the vapor becomes so high, that the rewetting liquid will be directly evaporated on the wall before it flows into
 152 the cavity. Then macro film boiling on the wall sets in. As in practice the cavity geometry is generally unknown,
 153 it is highly difficult to provide a mechanistic modelling for the heat transfer processes inside the cavity.
 154 However, the decisive point is, that $t_a = t_r$ is a significant feature of critical heat flux. As we will derive below,
 155 this criterion together with heat flux balances is sufficient to determine the onset of CHF without any further
 156 modelling of in-cavity processes. As at this point the system is still marginally stable, we consider the heat flux
 157 at this very moment as near critical heat flux CHF-. With Eq. (3) and $t_w = t_a = t_r$ it is given as

$$CHF = \frac{\left(\frac{4}{3}\pi r_d^3 \rho_g h_{fg}\right)}{\pi r_d^2 (t_g + t_c + t_r)}. \quad (4)$$

159 According to Zhao [28], under constant heat flux, the thermal layer recovery time is derived as

$$t_r = \left[\frac{\pi k_l}{2\dot{q}}\right]^2 \frac{(T_w - T_b)^2}{\pi \alpha_l}, \quad (5)$$

162 where \dot{q} is the feeding heat flux, T_w is the wall temperature, T_b is bulk temperature, k_l is the liquid thermal
 163 conductivity and α_l is the liquid thermal diffusivity. As stated earlier, the bubble growth rate in the inertia
 164 controlled period is much higher than for thermal diffusion. As inertial growth is quite fast, we may omit this
 165 period in further calculations and just consider the thermal diffusion controlled growth, giving $t_g = \left(\frac{r_d}{B}\right)^2$ from

167 Mikic [29], where B is a factor for evaporation written as $B = \left(\frac{12}{\pi} \alpha_l\right)^{\frac{1}{2}} \frac{\rho_l c_{pl} (T_w - T_{sat})}{\rho_g h_{fg}}$. Thereby, T_{sat} is
 168 saturation temperature and c_{pl} is liquid specific heat capacity. In the high subcooling case, we may assume that
 169 the condensation on the bubble also follows the rule of growth but opposite to evaporation with a factor B_c
 170 with subcooling $T_b - T_{sat}$. Then the condensation time t_c can be calculated with t_g as

$$t_c = \left(\frac{r_d}{B_c}\right)^2 = t_g \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2}. \quad (6)$$

172

173 Note, that we have readily assumed that the gas in the bubble and the condensed liquid around it is always at
 174 saturation temperature (Figure 3c). The same holds for the rewetting liquid in low subcooling, as this is a
 175 mixture of the liquid surrounding the bubble. To keep notation less complex we will in the following introduce

176 the latent heat $H_s = \rho_g h_{fg}$ and the parameter $X = \left(1 + \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2}\right)$. r_d is the departure radius, which is in

177 some cases dominated by thermal properties and in other cases by the shear stress.

178 When r_d is dominated by local conditions and thermal properties, the critical departure diameter is defined as
 179

$$r_{d,c} = \frac{2}{3} \left(\frac{1}{2} \cdot \frac{(T_w - T_{sat})}{(T_w - T_b)} + 1 \right) \cdot \frac{((T_w - T_{sat}))}{\left(\frac{(T_{sat} - T_b)}{2}\right)} C^* \sqrt{(\sigma/g\Delta\rho)} \left(\left(\frac{12}{\pi} \alpha_l\right)^{\frac{1}{2}} Ja^* \right)^n, \quad (7)$$

180

181 where $Ja^* = \frac{\rho_l c_{pl}(T_{sat})}{\rho_g h_{fg}}$, $C^* = 1.09e - 7$ and $n = 1.7$ for water [30-31]. The calibration process itself is
 182 described in [25].

183 When r_d is dominated by the shear stress, the critical departure diameter is defined as

184

$$r_s = \frac{C_2 \cdot 2\sigma k_l}{\mu_l h_c u_{max} Pr^{1/3}} = \frac{0.000364 \cdot \left(\frac{D}{D_0}\right)^{0.35} \left(\frac{L}{L_0}\right)^{0.25} \left(\frac{P}{P_0}\right)^{-2.9} \left(\frac{G}{G_0}\right)^{1.15} f(\Delta T_b)}{h_{c,s} u_{max}} \cdot \frac{2\sigma k_l}{\mu_l Pr^{1/3}} \quad (8)$$

185

186 where σ is the surface tension, $u_{max} = \frac{120}{98} \bar{u}$ is the maximum velocity, and \bar{u} is the average velocity in the
 187 flow channel. $h_{c,s}$ is heat transfer coefficient for single phase [32] defined via a Dittus-Boelter type equation as
 188

$$h_{c,s} = 0.023 \frac{k_l}{D} \left(\frac{DG}{\mu_l}\right)^{0.8} \left(\frac{\mu_l c_{p,l}}{k_l}\right)^{0.4}, \quad (9)$$

189

190 D is the diameter of pipe, and G is the mass flux, and μ_l is liquid viscosity. Thereby $f(T_b)$ is a function of the
 191 bulk temperature T_b , which is given as

192

$$f(T_b) = \begin{cases} \left(\frac{\Delta T_{b,0} [K]}{\Delta T_b} \cdot \frac{h_{c,s}}{h_{c,s,0}}\right)^{0.25} & \Delta T_b > 26 [K] \cdot \frac{h_{c,s}}{h_{c,s,0}} = 7.32 \times 10^{-4} [K^2 m^2 W^{-1}] h_{c,s} \\ 1 & \Delta T_b \leq 7.32 \times 10^{-4} [K^2 m^2 W^{-1}] h_{c,s} \end{cases} \quad (10)$$

193

194 $h_{c,s,0} = 3.55 \times 10^4 \text{ W m}^{-2} \text{ K}^{-1}$ is the single phase heat transfer coefficient at the reference point calibrated
 195 from Bergles' experiments. In flow boiling the bubble will be inclined as a truncated spherical bubble. We may
 196 assume the surface tension force in the wall tangential direction is C_2 times of that in the wall perpendicular
 197 direction of a regular spherical bubble. In the original paper, it is found to be $C_2 = 3.64e - 4$ for water. CHF
 198 usually happens randomly without any schematic order at the downstream part near the end or just at the end of
 199 the pipe or channel [33-35]. In our case, for simplification, we consider that CHF always happens at the end of
 200 the pipe. The excess bulk temperature measured in the Bergles' case is ΔT_b in the calculation of r_s .

201 Then Eq. (4) yields

202

$$CHF = \frac{B^2}{2 \cdot \min(Xr_{d,c}, r_s)} \left[\frac{4}{3} H_s - \sqrt{\left(\frac{4}{3} H_s\right)^2 - \frac{X(k_l(T_w - T_{sat}))^2 \pi}{\alpha_l B^2}} \right]. \quad (11)$$

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This particular dependency on $Xr_{d,c}$ and r_s leads to the interpretation that CHF may occur as two different phenomena. One is a local CHF when $Xr_{d,c}$ dominates. The CHF- will then be based on local conditions and independent of the void fraction in the bulk. It usually occurs in pool boiling or forced convective boiling with low mass flux. The other is a global CHF when r_s dominates. Because r_s is determined by local shear stress, CHF will be impacted by the void fraction in the bulk, which is dependent on the upstream conditions and also the diameter and length of the channel. It occurs in forced convective boiling with high mass flux. The concept and model was already validated with six different experiments covering different operational conditions such as pressure, mass flux, subcooling, wall surface properties, channel length and diameter and so on in pool and flow boiling process with water [25].

213 2.1 Extension of the CHF- concept

214 The $r_{d,c}$ in the CHF- model was derived based on the formula of Cole and Rohsenow [36]
215

$$r_d = \frac{1}{2} C \left(\sigma / g (\rho_l - \rho_g) \right)^{\frac{1}{2}} \left(\rho_f c_{pl} T_{sat} / \rho_g h_{fg} \right)^{1.25} = C \left(\sigma / g (\Delta \rho) \right)^{\frac{1}{2}} (Ja^*)^{1.25} \quad (12)$$

216 and further considers the mutual effect raised by Kolev in 1994 [37].
217 The mutual effect can be described as an average fluctuation velocity, which enhances an additional shear stress
218 and leads to earlier bubble departure. It is given as
219
220

$$\bar{V} = 2 \bar{A}^2 \bar{\tau}_g (\pi N)^{1/2} = 2 \bar{A}^2 \bar{\tau}_g \frac{1}{r_{d,c}} \approx \bar{A} = \left(\frac{\pi h_{fg} \rho_g \overline{\Delta T_{sup}}}{7 \rho_l T_{sat}} \right)^{\frac{1}{2}} \propto \left(\frac{\Delta T_{sup}}{Ja^*} \right)^{\frac{1}{2}}. \quad (13)$$

221 As is stated in Eq. (13), $\overline{\Delta T_{sup}}$ and T_{sat} impacts the fluctuation velocity \bar{V} and further the critical departure
222 radius $r_{d,c}$. In our recent analysis [25] we employed the experimental data from Sakashita [30], [31] to derive
223 $r_{d,c}$ for pool boiling. As stated by Sugrue et al. [38], the pressure increase from 1 bar to 5 bar leads to 52%
224 higher saturate temperature but no considerable change of $\overline{\Delta T_{sup}}$. Hence, at that time, we neglected $\overline{\Delta T_{sup}}$
225 in the derivation of $r_{d,c}$. That is, Eq. (7) comes without $\overline{\Delta T_{sup}}$ while the dependency of $r_{d,c}$ on T_{sat} or Ja^* is
226 given. Comparing Eq. (7) and (12), the impact of Ja^* on the change from r_d to $r_{d,c}$ via \bar{V} can be defined as
227
228

$$\frac{r_{d,c}}{r_d} \propto B^{*(1.7-1.25)} = B^{*0.45} \propto Ja^{*0.45}. \quad (14)$$

229 Now, we would like to extend the model towards low mass flux flow boiling at a constant pressure. According
230 to Chen 1966 [32] the heat transfer coefficient is given as
231
232

$$h_c = 0.023 \frac{k_l}{D} \left(\frac{DG(1-\varphi)}{\mu_l} \right)^{0.8} \left(\frac{\mu_l C_{p,l}}{k_l} \right)^{0.4} \cdot F = \frac{\dot{q}}{\Delta T_{sup} + \Delta T_{sub}}. \quad (15)$$

233 Here, φ is the weight fraction of vapour and F is a function of the two-phase pressure drop [32]. As h_c depends
234 on $\overline{\Delta T_{sup}}$ and G , we must further consider this dependency.
235
236
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238 From Eq. (13) we see, that the impact of $\overline{\Delta T_{sup}}$ on $r_{d,c}$ should be similar to that of $(\frac{1}{Ja^*})$, which is further
 239 quantified in Eq. (14) and gives
 240

$$\frac{r_{d,c}}{r_d} \propto (\overline{\Delta T_{sup}})^{-0.45}. \quad (16)$$

241 Because r_d is not a function of $\overline{\Delta T_{sup}}$, this dependency can be simplified to
 242
 243

$$r_{d,c} \propto (\overline{\Delta T_{sup}})^{-0.45}. \quad (17)$$

244 If we have a calibrated reference value $r_{d,c,0}$, Eq. (16) can be rewritten as
 245
 246

$$\frac{r_{d,c}}{r_{d,c,0}} \propto \left(\frac{\overline{\Delta T_{sup}}}{\overline{\Delta T_{sup,0}}} \right)^{-0.45}. \quad (18)$$

247 If $x = \frac{\overline{\Delta T_{sub}}}{\overline{\Delta T_{sup}}}$, then the Eq. (15) can be rewritten as
 248

$$\overline{\Delta T_{sup}}(1+x) = \frac{\dot{q}}{h_c}. \quad (19)$$

249 Then for the same heat flux, Eq. (15) yields
 250

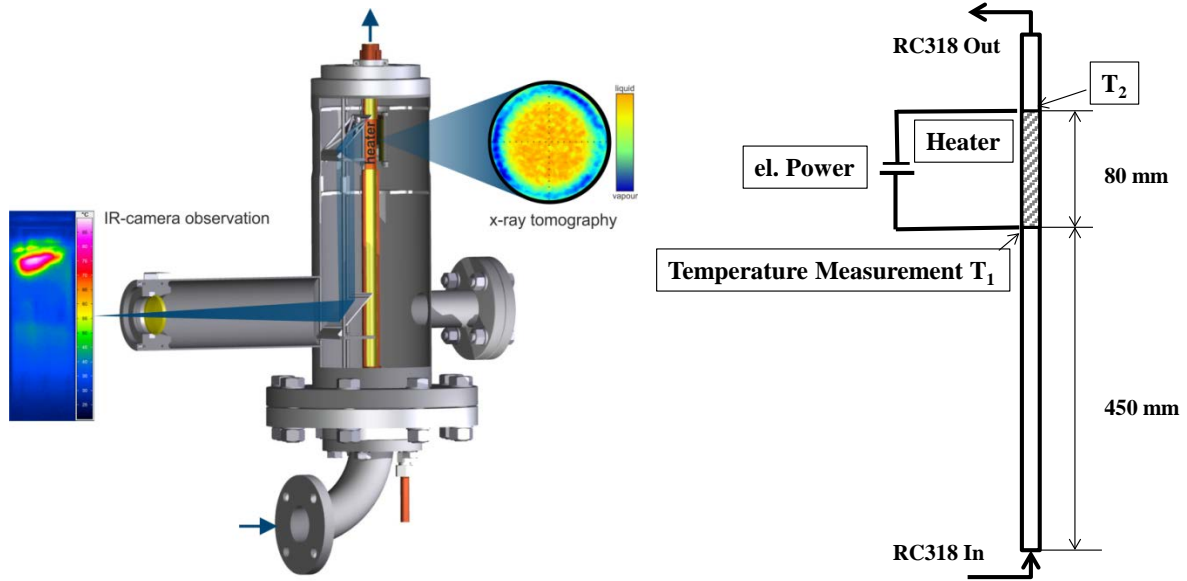
$$\frac{r_{d,c}}{r_{d,c,0}} \propto \left(\frac{h_{c,0,i}(1+x_i)}{h_{c,0,0}(1+x_0)} \right)^{0.45}. \quad (20)$$

251 In order to simplify the calculation, we will further assume that x_i is always approximately equal to x_0 in this
 252 low subcooling case.

253 3. Experiment

254 An experimental setup has been devised, which allows investigating the boiling heat transfer and local
 255 development of CHF. The setup (Figure 4) is a flow loop operated with the refrigerant Octafluorocyclobutane
 256 (RC318) (boiling point 25°C at 0.315 MPa pressure, $\rho_g=35.7 \text{ kg/m}^3$, $\rho_l=1365 \text{ kg/m}^3$, $c_{pl}=1130 \text{ J/(kgK)}$,
 257 $k_l=0.063 \text{ W/(mK)}$, $h_{fg}=100,560 \text{ J/kg}$). Due to the lower boiling point compared to water, the experiment
 258 facility with RC318 is easier to operate and control. The heated test section is an 80 mm long vertical stainless
 259 steel tube (outer diameter 20 mm, thickness 0.3 mm) axially connected to massive copper conductor tubes. The
 260 distance between the inlet to the heated section is 450 mm to ensure a fully developed flow. The refrigerant loop
 261 comprises a pump for conveying the liquid, two plate heat exchangers for adjusting sub-cooling temperatures
 262 and removing the heat, and an expansion vessel for safety reasons in case of an uncontrolled burn-down of the
 263 heater. Additional auxiliary systems (refrigerant maintenance pump, vacuum pump, storage vessel) are available
 264 for refrigerant handling.

265 The facility is equipped with an infrared camera (Infratec VarioCam HD) and gold mirrors to measure wall
 266 temperatures at the test section with a nearly full 360° angular coverage. The infrared camera has a resolution of
 267 640x480 pixels at a frequency of 60 Hz. A special fast camera-signal based safety shut-down mechanism
 268 prevents heater destruction when boiling crisis is reached. The whole facility is combined with a fast X-ray
 269 tomography scanner system [39], which has, however, not been used in this study.

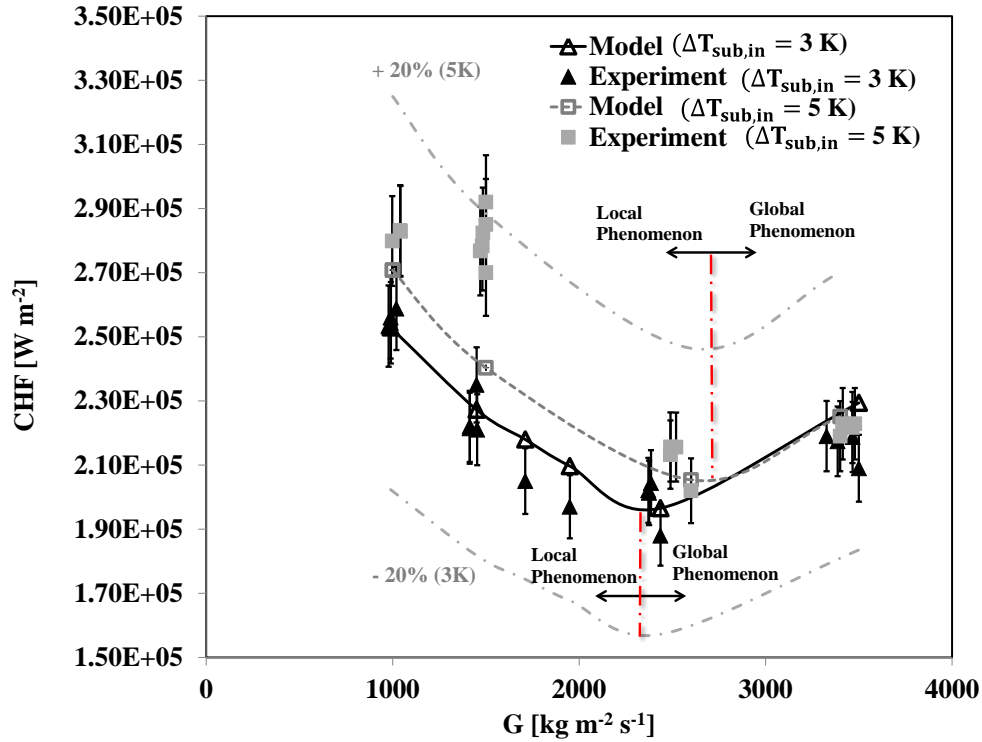


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271 *Figure 4: Multi Geometry Critical Heat Flux Observation facility (MORENA).*

272 Experiments were run at different mass fluxes. For each mass flux we started heating at a total rate of 62% of
 273 CHF and increased the heating power in 6 to 7 steps with a gradually decreased power increment from 9% to
 274 2.5% of CHF until CHF detection. Before each stepwise power increase, we waited until the wall superheat was
 275 in equilibrium, which could be derived from a distributed wall temperature measurement with a Fiber Bragg
 276 Grating sensor [40]. The experimental standard deviation in the measured CHF values was determined as 2.4%
 277 from multiple tests.

278 **4. Results and Discussion**

279 The CHF- concept [25] requires calibration of the boiling onset, which accounts for the wall material and surface
 280 characteristics. The onset is calibrated with the case of $G = 990 \text{ kg m}^{-2} \text{ s}^{-1}$ and $\Delta T_{sub,in} = 3.3 \text{ K}$. The onset wall
 281 superheat is 2.64 K. Though the applied liquid in the experiment is RC318, we found that the value of C_2 in the
 282 calculation of r_s is approximately equal to that of water ($C_2 = 3.64e - 4$). The model with these two calibrated
 283 constants has been further applied to predict the CHF under all conditions.



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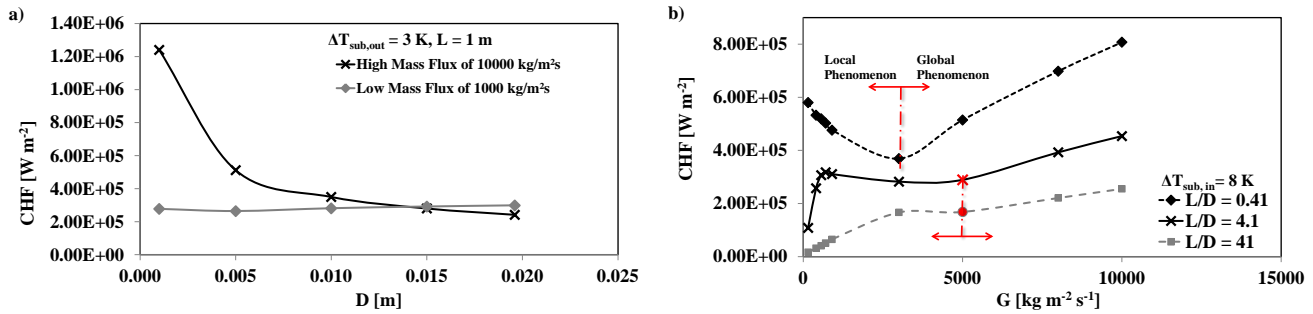
Figure 5: Comparison of experimental and calculated critical heat flux versus mass flux at ~ 3K and ~ 5 K inlet subcooling.

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The average error between experiment and model is around 5.3%. As shown in Figure 5, CHF decreases and then increases with an increase of mass flux. The critical mass flux for 3 K subcooling is around $2400 \text{ kg m}^{-2} \text{ s}^{-1}$. That is, when the mass flux is less than $2400 \text{ kg m}^{-2} \text{ s}^{-1}$, the CHF is initiated with the mechanism corresponding to $r_{d,c}$, that is, CHF is locally initiated. When the mass flux is larger than $2400 \text{ kg m}^{-2} \text{ s}^{-1}$, the CHF is initiated with the mechanism corresponding to r_s , which is dependent on the void fraction and local flow pattern. In this case, CHF is globally initiated. Further, with an increase of the subcooling temperature from 3 K to 5 K, the critical mass flux increases from $\sim 2400 \text{ kg m}^{-2} \text{ s}^{-1}$ to $\sim 2800 \text{ kg m}^{-2} \text{ s}^{-1}$. The experimental results confirm the hypothesis of two initiation mechanisms from the theoretical CHF- concept.

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As channel orientation, channel length and hydraulic diameter are known to have an impact on CHF we applied the CHF- concept to explain previous experimental findings qualitatively. At high mass flux the bubble departure is dominated by shear stress (Eq. (8)), no matter, if orientation is vertical or horizontal. But for pool boiling and low mass flux flow boiling, buoyancy plays a role (as modelled by the term $g(\Delta\rho)$ in Eq. (7)). Buoyancy is impacted by the orientation angle. That is, in low mass flux cases, CHF can still depend on orientation angle (Celata and Mariani, [2]). This conclusion holds e.g. for the studies of Papell et al 1966 [5] on the effect of buoyancy at different mass flux conditions. Further we refer to the investigation of the impact of L and D by Nariai and Insaka [4] to assess the role of outlet subcooling. For a similar outlet subcooling and local CHF channel length L and channel diameter D have an effect on CHF via the average heat transfer coefficient (Eqs. (7), (9) and (20)). This impact is, however, not very significant (Figure 6a). But CHF will be significantly impacted by L, D and L/D ratio at high mass flux when CHF is globally initiated, because there CHF is directly dependent on L and D (Eq. (8)). This conclusion agrees well with the experimental finding of Nariai and Insaka in 1992.



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 310 *Figure 6: Calculated critical heat flux for a) different pipe diameters (L/D ranges from 50 to 1000) at high and*
 311 *low mass flux and constant outlet subcooling of 3 K and b) different mass flux with different L/D ratio ($D = 0.02$*
 312 *m) at constant inlet subcooling of 8 K.*

313 Further we extended our study to consider a larger mass flux range (150 to 10000 $\text{kg}/(\text{m}^2\text{s})$) for different L/D
 314 ratios (Figure 6b). Different to the Figure 6 a), the subcooling at inlet is the condition of modeling in Figure 6 b)
 315 As the natural circulation velocity is usually is around 0.1 m/s, we set the lowest mass flux to 150 $\text{kg}/(\text{m}^2\text{s})$.
 316 Further we did the CHF calculation with a constant inlet subcooling of 8 K. The simulation results indicate that
 317 the impact of mass flux on CHF differs for different L/D ratio. For small L/D ratios an increase of mass flux
 318 leads to a decrease of CHF when it is locally initiated. When the L/D ratio becomes intermediate, CHF increases
 319 firstly and then decreases until it turns into a global phenomenon. Further when the L/D ratio is large enough,
 320 the increase of mass flux will not lead to a decrease of CHF even when it is locally initiated. When CHF is a
 321 globally initiated, the CHF always increases with an increase of mass flux.

322 CHF at low pressure and low velocity (LPLV) conditions is important for nuclear reactors in accident conditions
 323 as well as small district heating reactors [12]. Therefore it has been widely investigated for several decades. But
 324 it was still not well predicted by the present models or correlations which were made for high pressure and high
 325 velocity (HPHV) conditions [41]. Chang et al. [12] conducted an experiment at 1 bar with water at low mass
 326 flux in a pipe of $L/D = 82$. They found that the measured zero-flow CHF (30 – 40 kW/m^2) is much lower than
 327 the one predicted by an ordinary pooling boiling CHF correlation ($\sim 1000 \text{ kW}/\text{m}^2$). They conjectured that the
 328 possible CHF mechanisms under LPLV conditions are flooding, churn to annular flow transition, dryout of
 329 liquid film in slug flow and dryout of liquid film in annular flow, which leads to an overestimation of the CHF
 330 value by the present mechanistic models or correlations generated for HPHV conditions. From this work, we
 331 find the missing chain between the CHF at low mass flux conditions and CHF in pool boiling. At smaller L/D
 332 ratio, CHF at lowest mass flux becomes larger. We believe that it will approach to the pool boiling CHF value
 333 when L/D decreases to a relatively low level. We may further conclude that the overestimated prediction of the
 334 CHF at LPLV conditions by common models or correlations may be just because of different CHF initiation
 335 mechanisms, which we have now introduced in the CHF- concept.

336

337 5. Conclusion

338 There are two initiation mechanisms of CHF derived from the CHF- concept. One is dependent on the local
 339 conditions. In this case, CHF is a local phenomenon. The other is dependent on the void fraction in the bulk that
 340 is a global effect from upstream. Then, CHF is a global phenomenon. In this work, these two different
 341 mechanisms and phenomena were clearly confirmed by experiments at different mass fluxes and subcooling.
 342 The two occurrences of CHF and the corresponding mechanisms explain different impacts of variables such as
 343 channel orientation angle, buoyancy and L/D at high and low mass flux, which were already found in earlier
 344 investigations. Further it also explains the failure of CHF prediction at low pressure and low velocity (LPLV)
 345 conditions with common mechanistic models and correlations.

346 Further it was found that the impact of mass flux on CHF for the two different phenomena is different. When
 347 CHF is local, it CHF decreases with an increase of mass flux at a small L/D ratio. At an intermediate L/D ratio,
 348 CHF increases firstly and then decreases with the increase of mass flux. Further it increases with the increase of
 349 mass flux at a high L/D ratio. In the high mass flux range, when CHF is a global phenomenon, it increases with

350 the increase of mass flux, as was already observed by other researchers. The CHF- concept seems to provide the
 351 missing modelling link between pool boiling and zero velocity flow boiling.

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356 **Nomenclature**

357	A	a parameter defined as $\frac{\pi h_{fg} \rho_g \Delta T_{sup}}{7 \rho_l T_{sat}}$
358	A_b	projective area of bubble defined as πr_d^2
359	B	a parameter defined as $\left(\frac{12}{\pi} \alpha_l\right)^{\frac{1}{2}} Ja$
360	B^*	a parameter defined as $\left(\frac{12}{\pi} \alpha_l\right)^{\frac{1}{2}} Ja^*$
361	B_c	a parameter defined as $\left(\frac{12}{\pi} \alpha_l\right)^{\frac{1}{2}} \rho_l c_{pl} (T_b - T_{sat}) / \rho_g h_{fg}$
362	c_{pl}	specific heat capacity of liquid
363	C^*, C_2	constant or parameter
364	Fr	modified Froude number
365	D	hydraulic diameter
366	G	mass flux
367	g	gravity
368	H_s	a parameter defined as $\rho_g h_{fg}$
369	h_c	heat transfer coefficient
370	$h_{c,s}$	heat transfer coefficient for single liquid phase
371	h_{fg}	latent heat
372	Ja	Jakob number
373	Ja^*	modified Jakob number $\rho_l c_{pl} T_{sat} / \rho_g h_{fg}$
374	k_l	liquid thermal conductivity
375	L	length of heated wall
376	N	nucleation site density
377	n	constant
378	p	pressure
379	Pr	Prandtl number
380	\dot{q}	heat flux
381	\dot{q}_{CHF-}	critical heat flux
382	Q	heat of bubble life cycle
383	Q_b	heat in the bubble
384	$Q_{b,c}$	heat due to condensation around the bubble
385	$Q_{b,s}$	heat flowing from superheated liquid surrounding the bubble
386	$Q_{b,w}$	heat of bubble from wall
387	Q_q	quenching heat
388	r	radius
389	r_c	critical radius on the mouth of cavity
390	r_d	bubble departure radius

391	$r_{d,c}$	critical departure radius due to local conditions
392	r_s	critical departure radius due to shear stress
393	T_b	bulk temperature
394	T_w	wall temperature
395	T_{sat}	saturation temperature
396	ΔT_{sup}	superheat
397	$\overline{\Delta T}_{sup}$	averaged wall superheat
398	ΔT_{sub}	subcooling
399	t	time
400	t_a	cavity activation time
401	t_c	bubble condensation time
402	t_g	bubble growth time
403	t_r	thermal layer recovery time
404	t_w	bubble waiting time
405	τ_g	inertial growth time
406	\overline{V}	average fluctuation velocity
407	X	a parameter defined as $1 + \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2}$
408	x	a parameter defined as $\frac{\Delta T_{sub}}{\Delta T_{sup}}$
409	α_l	liquid thermal diffusivity
410	ϕ	orientation angle
411	σ	surface tension
412	μ	viscosity of liquid
413	ρ_g	density of vapor
414	ρ_l	density of liquid
415	$\Delta\rho$	density difference of vapor and liquid

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