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Quantitative prediction of critical heat flux initiation in pool and flow boiling

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Abstract

Boiling is a very efficient heat transfer mechanism with a large heat transfer coefficient and it is widely found in industrial systems. However, boiling heat transfer is limited by the critical heat flux (CHF), also termed as boiling crisis. It leads to a rapid decrease of the heat transfer coefficient in temperature controlled heat transfer or to a significant jump in heater surface temperature in power controlled heat transfer cases. While the earlier effect clearly lowers efficiency the latter may even jeopardize safety. A clear understanding of the basic mechanisms leading to CHF is still lacking. In this paper a new model of priori critical heat flux(CHF-) is derived from the bubble dynamics of nucleate boiling. It holds for pool boiling and forced convective boiling and incorporates a mutual effect model and a shear stress model. The comparison between predicted and experimental results under different thermal hydraulic conditions shows a good agreement. The model is capable to explain the initiating mechanism of the boiling crisis and impacts from different variables. It can be also implemented as a sub-model in CFD codes.

Keywords: Critical heat flux (CHF), boiling heat transfer, pool boiling, forced convective boiling, cavity activation

1. Introduction

Depending on the wetted surface fraction, boiling can occur in three modes: (partial and fully developed) nucleation boiling, transition boiling and film boiling [1]. Nucleation boiling is most efficient and associated with a very large heat transfer coefficient due to two effects: frequent wetting and de-wetting of the wall by liquid and convective transfer of steam into the bulk fluid by departing bubbles. Transition boiling occurs from the critical heat flux point (CHF) where part of the surface starts to be irreversibly covered by vapor. From then on, the heat transfer coefficient begins to decrease for temperature controlled heat transfer under unstable conditions until all the heated wall is fully covered by vapor. This is then referred to as film boiling. For power controlled heating, a sudden drop of the heat transfer coefficient leads to a rapid increase of wall temperature, which potentially leads to heater meltdown. Understanding and predicting the complex phenomena involved in CHF is necessary for the efficient and safe operation of industrial heat transfer systems, like boilers, nuclear reactors, or electronics/microchips cooling systems. In the last decades, the mechanism for the transition from nucleation boiling to CHF and further to film boiling has been widely investigated. Different system variables affecting the CHF were already identified and analyzed: pressure [2-5], local liquid subcooling [2, 6, 7, 8], mass load (in subcooled flow boiling) [9-12], heated wall length, hydraulic diameter (in subcooled flow boiling) [13, 14], wettability, roughness and porosity [15, 16, 17]. Further different theoretical models to describe the CHF, such as the Hydrodynamic Instability Model [18-22], the Near-Wall Bubble Crowding Model [23, 24], the Liquid Sublayer Dryout Model [25-28], Bubble interaction theory [29] and others [35] were also developed and compared with experiments. The most widely accepted CHF model are two hydrodynamics instability model at present: the hydrodynamics instability model proposed by Zuber [21] and Liquid Sublayer Dryout model proposed by Haramura

44 and Katto [26]. The hypothesis of the Zuber's model is that the down flow of fresh liquid to the heat
45 surface is prevented by the upward flow of vapor due to the Helmholtz instability. According to
46 Haramura and Katto's model, the CHF is also a result of the Helmholtz instability, the columnar
47 structure of vapor stems collapses with a vapor film blanketing a thin liquid film on the heater surface.
48 These models are widely recognized and validated with experimental results. However if CHF is only
49 due to hydrodynamics, it is difficult to explain the influence of the heating wall conditions (roughness,
50 wettability, thickness, material and so on) on CHF. The other problem of the present CHF models is
51 that the occurrence of the burnout is always treated as independent of the nucleate boiling process.
52 Sadasivan et al. [30] concluded that due to CHF occurs as the upper limit of the nucleate boiling
53 region, it is reasonable to expect that the different physical phenomena involved in the nucleate boiling
54 region should interact such that CHF value. A realistic CHF model would be one that is a natural
55 outcome of the description of the high heat flux nucleate boiling region. In 2009, Kolev [29] tried to
56 build a theory to connect the nucleation boiling and CHF. He includes the effect of the shear force
57 generated by mutual interaction of growing and departing bubbles. This shear stress shortens the
58 bubble life cycle, reducing the bubble departure diameter which reduces the latent heat removal per
59 bubble cycle. When this effect becomes dominant, the CHF is approached. However, the influence of
60 the heating surface is still missing in this model except contact angle (wettability).
61 Today, the assessment of system designs with respect to promotion or prevention of CHF is still based
62 on expensive experiments. Many correlations developed from such experiments have been applied in
63 some specific 1D codes for engineering design. However, these correlations are only valid in a limited
64 scope of applications. Computational fluid dynamics (CFD) is an attractive way to support engineering
65 design by 3D flow simulation in the future. It would be beneficial, if occurrence of CHF could be
66 simulated with CFD codes. However this requires a CHF model which can clearly explain the CHF
67 initiation mechanism from nucleation boiling. Consequently, a successful CHF model should at least:

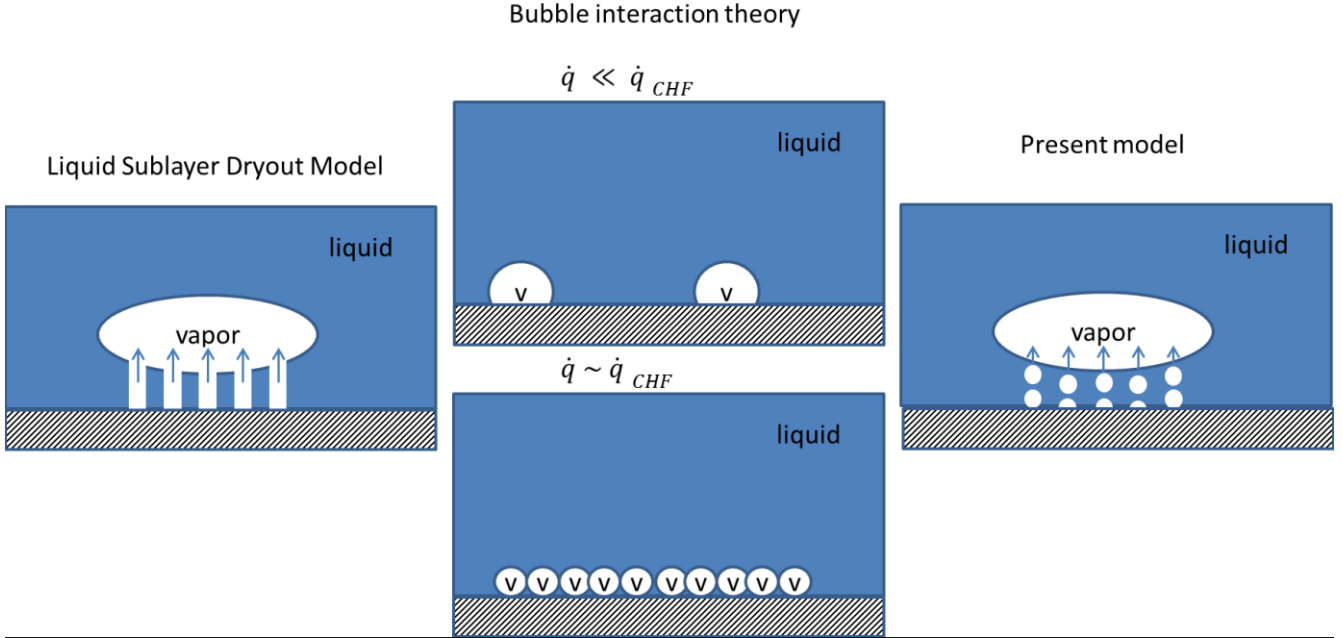
- 68
- 69 a) be able to consider both pool boiling and flow boiling;
- 70 b) be time and position dependent to explain the local wall temperature fluctuation;
- 71 c) be able to consider the effects of wall superheat and the CHF initiation mechanism;
- 72 d) cover the effects of the surface parameters;
- 73 e) be suitable to be implemented in a CFD code.

74

75 These criteria were also partly mentioned in Zhao's work [32]. In this study we analyzed the complex
76 mechanisms of cavity activation and heat transfer in the nucleate boiling process. From this analysis, a
77 model of priori critical heat flux (CHF-) is inferred. Further this model is developed into formulae for
78 pool and forced convective boiling. The calculation results are compared with experimental results
79 from different experiments for validation.

80 This work applies part of idea from bubble interaction theory. Instead of pure mutual effect of bubbles,
81 we pay attention on the thermal effect during nucleation boiling on/in the wall. We also considered the
82 mushroom structure of bubbles appear near the CHF which is well observed by experiments but with
83 columnar of isolated bubble between the mushroom shaped bubble instead of the stem introduced by
84 Liquid Sublayer Dryout model. Different to total sublayer dryout mechanism, we consider the
85 columnar of bubbles dominate the CHF. This work doesn't contrary to the previous founding such as
86 the irreversible dry spot in experiments [31, 37], because we pay attention on the priori critical heat
87 flux where the system still have stable bubble generation when the wall temperature starts to climb.
88 When the temperature continues climbing up to certain level like introduced ~134 °C in the experiment
89 from Kim [31] (water horizontal pool boiling at 1 atm), the irreversible dry spot will be formed. In the
90 other word, lower than this temperature, the wall surface still has chance to be rewetted. In this work,

91 we try to explain why even the rewetting does not stop the temperature climbing until irreversible dry
 92 out is formed and CHF is approached.



93
 94 *Figure 1: Typical Liquid Sublayer Dryout Model, bubble interaction theory and new estimated bubble*
 95 *nucleation structure under mushroom shaped bubble*

96 **2. Results and Discussion**

97 **2.1 Concept**

98 In this paper we fundamentally consider the bubble growth process in nucleation boiling as a stable and
 99 repeating process, which consists of cavity activation, bubble growth, bubble departure and associated
 100 surface rewetting. This concept is widely accepted and has been described in many other papers [24,
 101 36, 48]. In the following we will derive our CHF- model by considering in detail characteristic
 102 durations, heat fluxes, and temperatures of wall, steam and bulk liquid for the different phases in
 103 nucleate boiling. While our analysis incorporates some models developed by other researchers, the key
 104 novelty of our approach is that CHF is considered as being initialized from nucleation on/in the wall
 105 and dealing with the recovery of cavity activation and thermal layer.

106 A commonly accepted prerequisite of nucleate boiling is the existence of nucleation sites, which are
 107 assumed to be small micrometer size cavities in the wall. It is further assumed that in the period of
 108 rewetting after a bubble departure there is always a tiny amount of vapor remaining captured in the
 109 cavities. We need to note here, that this is a model only, but one which is strongly supported by
 110 observations [33- 35]. The model assumptions for activation of a bubble are as follows.

111 The gas pocket in a cavity is considered as the seed for the subsequent bubble growth. It is at pressure
 112

$$p = p_0 + p_s = p_0 + \frac{2\sigma}{r_c}. \quad (1)$$

113
 114 with p_0 being the pressure in the bulk liquid, p_s the Laplace pressure of the gas-liquid interface and
 115 r_c is the critical nucleus radius for bubble grows. The heater wall is superheated at temperature
 116 $T_{wall} = T_y(0) = T_{sat}(p_0) + \Delta T_{sup}$ and has an exponential temperature profile into the bulk liquid. In

117 the second stage, called bubble growth, the overheated gas pocket in the cavity is further fed by
 118 evaporating liquid from the superheated liquid in the thermal boundary layer. When the bubble is still
 119 small, its growth in diameter is quite fast and determined by the inertia of the liquid being displaced.
 120 Hence this period is referred to as inertia-controlled growth. As near-wall shear stress hinders
 121 displacement of liquid in the very vicinity of the wall, a small micrometer size layer of liquid remains
 122 at the wall underneath the bubble. It is referred to as micro-layer. As superheat is highest in this layer,
 123 it subsequently contributes a lot to evaporation and disappears with time. After a while the growth of
 124 bubble diameter becomes slower and it is no longer limited by liquid displacement but by evaporative
 125 heat flux and hence heat flux through the gas-liquid interface. This period is referred to as thermal
 126 diffusion controlled growth. The third stage of the bubble cycle is bubble departure from the wall,
 127 which may be preceded a sliding motion along the wall. Immediately after bubble departure liquid
 128 from the near wall region replaces the disappearing gas volume. This is the fourth phase, or quenching
 129 phase. As the replenishing liquid is on average cooler than the unaffected liquid portions near the wall
 130 it needs to be reheated such that the thermal boundary layer over the wall is restored. All the stages
 131 have certain durations. Most important is the total growth period t_g and the waiting time t_w between
 132 bubble departure and new activation.

133 The above description of the bubble cycle is state of the art. In the following we will further develop
 134 this concept by bringing effects in the wall around the cavity into play. In the following we will
 135 qualitatively describe our concept and in the next sections derive equations to quantitate the effects.
 136 Firstly we define the relevant heat fluxes. The total transferred heat Q_b during bubble growth consists
 137 of three parts: heat flowing from the wall into the bubble via evaporation $Q_{b,w}$, heat flowing from the
 138 superheated liquid near the wall into the bubble $Q_{b,s}$ and condensation heat loss at the upper part of
 139 the bubble $Q_{b,c}$, that is, heat flowing out of the bubble into the bulk liquid. The importance of this
 140 concept lies in the fact that the bubble growth is fed with heat from two sources, the wall and the
 141 thermal boundary layer, though we cannot say for the moment, how the share is quantitatively. After
 142 bubble departure, a waiting time is required to reform the nucleus in the cavity and to recover the
 143 thermal layer, that is, to recover the consumed heat in the bulk. During this period of quenching, the
 144 heat Q_q will be delivered from wall to the liquid. As during bubble growth the liquid in the vicinity of
 145 the bubble has gained the heat $Q_{b,c}$ and lost the heat $Q_{b,s}$, we may assume that $Q_q = Q_{b,s} - Q_{b,c}$.
 146 Moreover, some heat is needed to reheat the trapped liquid in the cavity from the wall. However, due to
 147 the small liquid volume in the cavity, this term can readily be neglected. The heat balance can then be
 148 written as

$$149 \quad Q_{b,w} + Q_q = Q_{b,w} + Q_{b,s} - Q_{b,c} = Q_b. \quad (2)$$

150 Further on we define the projective area $A_b = \pi r_d^2$ of a fully developed bubble with departure radius
 151 r_d as the apparent heat transfer area for boiling heat transfer per single bubble. In this work, we
 152 consider that before CHF is approached, the bubble is still able to complete its departure process on the
 153 wall and accumulate near the wall to form big bubbles or mushroom structure (See *Figure 1*). Due to
 154 the waiting time is quite small for large value of superheat [42] we consider the influenced factor of
 155 bubble projective area is 1 when CHF is approached. The total transferred heat Q_b is given by
 156 $\frac{4}{3}\pi r_d^3 \rho_g h_{fg}$ where ρ_g is density of vapor and h_{fg} is the latent heat of the fluid. The heat flux in this
 157 area during bubble growth in the low subcooling case (*Figure 2 b*) is then given as
 158
 159

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

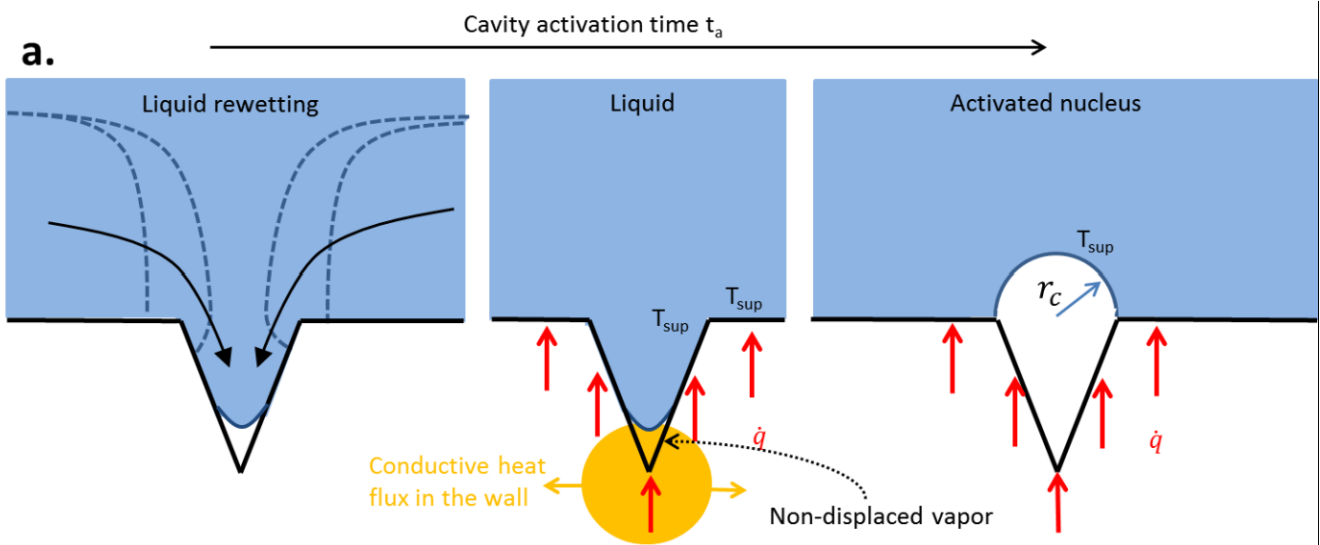
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where t_g is the growth time and t_w is the waiting time.

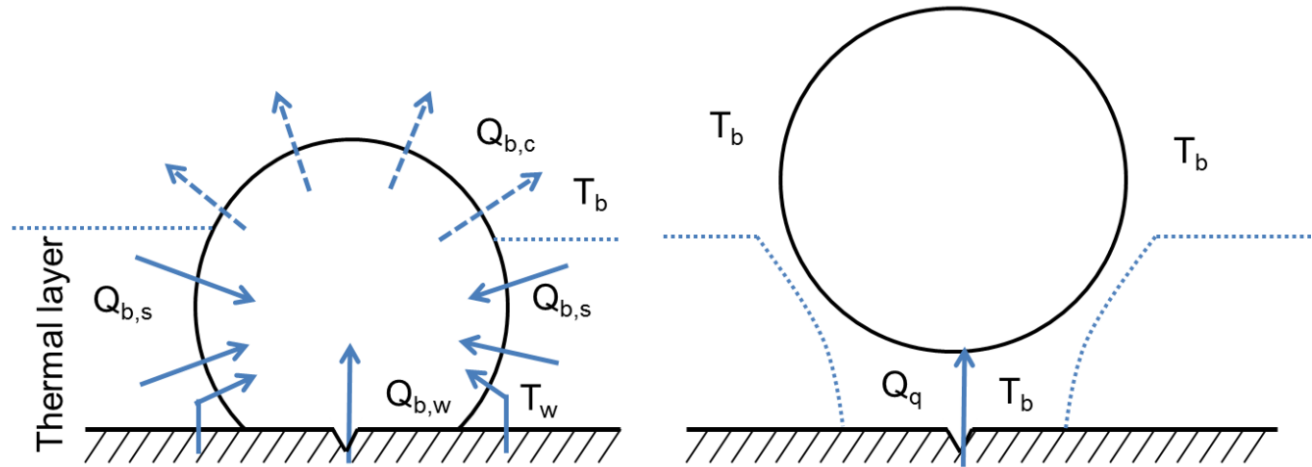
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b.

Condensation

Bubble Departure

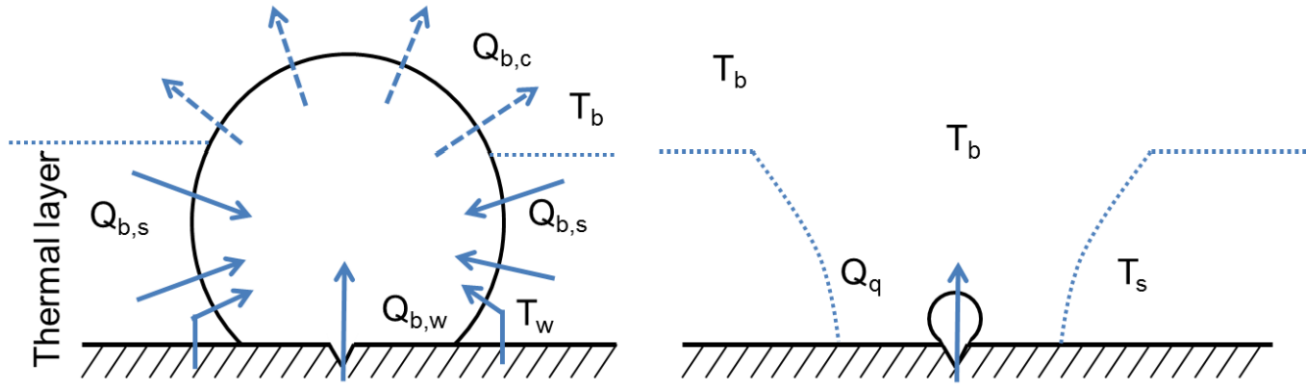


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C.

Condensation

Bubble shrinkage



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Figure 2: a) Cavity activation and heat transport during and after bubble growth: b) bubble departure in the low subcooling case and c) bubble shrinkage in the high subcooling and high heat flux case in horizontal pool boiling.

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Now, to complete the picture of nucleate boiling, we will further consider the case of strong subcooling (Figure 2 c). Then the bubble will not depart from the wall but condense when its cap penetrates into subcooled bulk liquid regions. Hence the bubble will activate, grow, shrink and collapse on the heated wall. For this case the heat flux is

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

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where t_c is the condensation time. For $t_c = 0$ Equation (4) equals Equation (3). During rewetting the replenishing liquid partially displaces the vapor in the cavity [32, 34]. Before the bubble cycle closes it takes a time period t_a (cavity activation time) to form a new vapor nucleus with critical radius r_c on the mouth of cavity, which is the activation condition. The formation of this vapor nucleus requires superheating of liquid in the cavity to overcome the high pressure in the nucleus. It is worthwhile to look at this process in more detail (Figure 2 a)). In the upper part of the cavity liquid touches the wall and we may assume that heating up this liquid has the same time scale as above the wall outside the cavity. In the lower part of the cavity there is vapor in direct contact with the cavity wall, which gives a low heat transfer coefficient. Hence we have a microscopic “film boiling” situation on the bottom of the cavity, which leads to a stronger superheat of the wall there. Conductive heat transfer towards the liquid through the wall material is then the major heat removal mechanism. The higher the heating power the higher becomes the wall superheat at the bottom of the cavity. On the other hand, as the vapor in the lower part of the cavity presents a strong heat transfer resistance, the liquid in the center of the cavity is heated slower than in single phase conditions and evaporation into the nucleus is delayed. This effect determines the activation time t_a . In our model this activation time plays a major role. As the vapor pocket delays the heating of the liquid in the cavity, we can assume that the recovery of the thermal boundary layer (recovery time t_r) occurs faster than the reactivation of the cavity, that is $t_a > t_r$. With increasing heating power or heat flux this situation can change. The wall material at the bottom of the cavity will superheat so much, that reheating of liquid in the cavity is again synchronous

194 with building of the thermal boundary layer outside the cavity, that is, $t_a = t_r$. If the heat flux further
 195 increases, the activation time becomes even shorter, the vapor nucleus grows rapidly into the yet not
 196 established thermal boundary layer and transfers additional heat to the boundary layer, then $t_a < t_r$.
 197 The larger one of t_r and t_a decides the total waiting time, that is, $t_w = \max(t_a, t_r)$. For $t_a < t_r$
 198 the situation is unstable, as increasing vapor content in the cavity will further increase cavity wall
 199 superheat and further shorten t_a . Eventually the temperature of the cavity wall and the vapor becomes
 200 so high, that the rewetting liquid will be directly evaporated on the wall before it flows into the cavity.
 201 Then macro film boiling on the wall sets in. As in practice the cavity geometry is generally unknown, it
 202 is highly difficult to provide a mechanistic modelling for the heat transfer processes inside the cavity.
 203 However, the decisive point is, that $t_a = t_r$ is a significant feature of critical heat flux. As we will
 204 derive below, this criterion together with heat flux balances is sufficient to determine the onset of CHF
 205 without any further modelling of in-cavity processes. As at this point the system is still marginally
 206 stable, we consider the heat flux at this very moment as priori critical heat flux **CHF-**. With Equation
 207 (4) and $t_w = t_r$ it is given as
 208

$$\dot{q}_{CHF-} = \frac{\left(\frac{4}{3}\pi r_d^3 \rho_g h_{fg}\right)}{\pi r_d^2 (t_g + t_c + t_r)}. \quad (5)$$

209

210 **2.2 Application of the CHF- model**

211 **2.2.1 Model of CHF- in Pool Boiling**

212 For pool boiling, according to Zhao [32], the wall temperature T_w surrounding the cavity, can be
 213 obtained from
 214

$$T_w = T_b + \frac{2\dot{q}\sqrt{t\alpha_l}}{\sqrt{\pi}k_l}, \quad (6)$$

215 where T_b is bulk temperature, k_l is the liquid thermal conductivity, α_l is the liquid thermal
 216 diffusivity. Here, \dot{q} is the feeding heat flux. For the recovery of the thermal layer, under constant heat
 217 flux, the thermal layer recovery time is derived as [32]

$$t_r = \left[\frac{\pi k_l}{2\dot{q}}\right]^2 \frac{(T_w - T_b)^2}{\pi \alpha_l}. \quad (7)$$

218 For a more detailed quantification of the bubble growth process we will now follow the concept of
 219 Mikic [38] developed in 1970. In this approach the bubble growth period is divided into two parts: 1)
 220 growth controlled by inertial forces and following that 2) growth controlled by the energy transfer
 221 through the vapour-liquid interface, referred to as thermal diffusion controlled period. From this
 222 assumption Mikic derived a general relation including both inertia and thermal diffusion controlled
 223 phenomena for the bubble growth rate in a dimensionless form given as
 224

$$r^+ = \frac{2}{3} \left[(t^+ + 1)^{\frac{3}{2}} - (t^+)^{\frac{3}{2}} - 1 \right] \quad (8)$$

225 with $r^+ = Ar/B^2$, $t^+ = A^2t/B^2$,

$$A = \left(\frac{\pi h_{fg} \rho_g \Delta T_{sup}}{7 \rho_l T_{sat}} \right)^{\frac{1}{2}} \quad (9)$$

$$B = \left(\frac{12}{\pi} \alpha_l \right)^{\frac{1}{2}} Ja \quad (10)$$

226 and the Jakob number

$$Ja = \frac{\rho_l c_{p,l} \Delta T_{sup}}{\rho_g h_{fg}}. \quad (11)$$

227 Here, $\Delta T_{sup} = T_w - T_{sat}$ is wall superheat, ρ_l is the liquid density, T_{sat} is the saturation
 228 temperature and $c_{p,l}$ is specific heat capacity of liquid. When t^+ is small ($t^+ \ll 1$) Equation (8) can
 229 be written as $r^+ = t^+$ or $r = At$ as a Rayleigh solution [38], which describes the growth under
 230 inertia control. For $t^+ \gg 1$, Equation (8) can be written as $r^+ = \sqrt{t^+}$ or $r = B\sqrt{t}$, which describes
 231 the growth in the thermal diffusion controlled period. As stated earlier, the bubble growth rate in the
 232 inertia controlled period is much higher than for thermal diffusion. As inertial growth is quite fast, we
 233 may omit this period in further calculations and just consider the thermal diffusion controlled growth,
 234 giving
 235

$$t_g = \left(\frac{r_d}{B} \right)^2. \quad (12)$$

236 In the high subcooling case, we may assume that the condensation on the bubble also follows the rule
 237 of growth but opposite to evaporation with a factor B_c , which is defined as
 238

$$B_c = \left(\frac{12}{\pi} \alpha_l \right)^{\frac{1}{2}} \frac{\rho_l c_{p,l} (T_b - T_{sat})}{\rho_g h_{fg}}, \quad (13)$$

239 and
 240

$$t_c = \left(\frac{r_d}{B_c} \right)^2 = t_g \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2}, \quad (14)$$

241
 242 The right-most term has been derived by combining Eqs. (10)-(13).
 243 Now Equation (5) yields

$$\dot{q} = \frac{Q_b}{\pi r_d^2 (t_g + t_c + t_w)} = \frac{\frac{4}{3} \pi r_d^3 \rho_g h_{fg}}{\pi r_d^2 \left(t_g \left[1 + \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2} \right] + t_w \right)}. \quad (15)$$

244 Note, that we have readily assumed that the gas in the bubble and the condensed liquid around it is
 245 always at saturation temperature (*Figure 2 c*). Then for low subcooling we can also assume, that the
 246 same holds for the rewetting liquid, as this is a mixture of the liquid surrounding bubble.

247 To keep notation less complex in the following we will introduce here the latent heat $H_s = \rho_g h_{fg}$ and
 248 the parameter $X = \left(1 + \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2}\right)$. Inserting Equation (7), Equation (12) and Equation (14) into
 249 Equation (15), we get
 250

$$\dot{q}_{CHF-} = \frac{B^2}{2Xr_d} \left[\frac{4}{3} H_s - \sqrt{\left(\frac{4}{3} H_s\right)^2 - \frac{X(k_l(T_w - T_{sat}))^2 \pi}{\alpha_l B^2}} \right]. \quad (16)$$

251 Equation (16) states that in horizontal pool boiling \dot{q}_{CHF-} is dependent on wall superheat and bulk
 252 temperature, i.e. \dot{q}_{CHF-} increases with increasing wall superheat under constant bulk temperature. The
 253 cavity with lowest activation superheat (boiling onset) has the lowest \dot{q}_{CHF-} on the wall where the
 254 boiling crisis should start. In the horizontal pool boiling, the departure diameter for saturated boiling
 255 maybe calculated by the Fritz equation [39] as
 256

$$D_{dp} = 0.0208\theta \left(\frac{\sigma}{g(\rho_l - \rho_g)} \right)^{1/2}, \quad (17)$$

257 where θ is liquid-solid contact angle, g is gravitational acceleration. Cole [40] further developed this
 258 formula and Cole and Rohsenow [41] derived an expression for the departure diameter by considering
 259 only the impact of differential pressure through Ja^* as
 260

$$D_{dp} = C \left(\frac{\sigma}{g(\rho_l - \rho_g)} \right)^{1/2} \left(\rho_f c_{pl} T_{sat} / \rho_g h_{fg} \right)^{5/4} = C \left(\frac{\sigma}{g(\Delta\rho)} \right)^{1/2} (Ja^*)^{5/4} \quad (18)$$

261 with $C = 1.5 \times 10^{-4}$ and $Ja^* = \rho_l c_{pl} T_{sat} / \rho_g h_{fg}$ for water. All the equations above are for single
 262 bubbles. In reality, when the boiling process is near CHF, the bubble nucleation is considered as fully
 263 developed, which means that a large amount of bubbles will be generated from the wall and the impact
 264 from neighbouring bubbles cannot be neglected. In 1994, Kolev [42] included the bubble mutual
 265 interaction into the calculation of bubble diameter. Due to the mutual effect, the bubble will depart or
 266 lift off earlier under the action of the shear force
 267

$$F_{d,t} = 0.3\pi\rho_l \left(2r_{d,c} \bar{V} \right)^2, \quad (19)$$

268 induced by the growth of neighbouring bubbles. Here, \bar{V} is the time averaged fluctuation velocity or
 269 micro-convection velocity. Under large heat flux near CHF, the average fluctuation velocity was given
 270 by Kolev as
 271

$$\bar{V} = B^2(\pi N)^{1/2}, \quad (20)$$

272
 273 with $N = \frac{1}{\pi r_{d,c}^2}$ being the local nucleation site density under the mushroom shaped bubble (See *Figure*
 274 *1*). Kolev already derived a solution to calculate the bubble departure diameter under different wall
 275 superheat [42]. However, we focus on the impact of the average mutual effect on the bubbles of the
 276 cavity activated at boiling onset, which this solution cannot account for. Though the inertia controlled

277 growth time is quite short and does not much influence the thermal transfers, it still dominates the
 278 average fluctuation velocity. With the average fluctuation velocity derived in Kolev's work, Equation
 279 (20) can be modified as

$$\bar{V} = 2\bar{A}^2\bar{\tau}_g(\pi N)^{1/2} = 2\bar{A}^2\bar{\tau}_g \frac{1}{r_{d,c}} \approx \bar{A} = \left(\frac{\pi h_{fg}\rho_g \overline{\Delta T_{sup}}}{7 \rho_l T_{sat}} \right)^{1/2} \propto \left(\frac{\overline{\Delta T_{sup}}}{Ja^*} \right)^{1/2}. \quad (21)$$

280 Here, $\bar{\tau}_g$ is the average bubble growth time. As the activation temperature for two neighboring
 281 cavities may be different, the averaged wall superheat $\overline{\Delta T_{sup}}$ is used here to calculate the average
 282 fluctuation velocity. We consider that the wall superheat has much less impact to the mutual effect than
 283 Ja^* or density and saturation temperature under different pressure. For example, Sugrue [55] reported
 284 under same other conditions, from 1 bar to 5 bar, the $\overline{\Delta T_{sup}}$ has nearly no considerable change while
 285 vapor density increases 5 times and saturate temperature increases 52%. As both mutual effect and
 286 departure diameter are functions of Ja^* , we estimate the bubble departure diameter near the CHF $r_{d,c}$
 287 as
 288

$$r_{d,c} = C^* \left(\sigma/g\Delta\rho \right)^{1/2} Ja^{*n}. \quad (22)$$

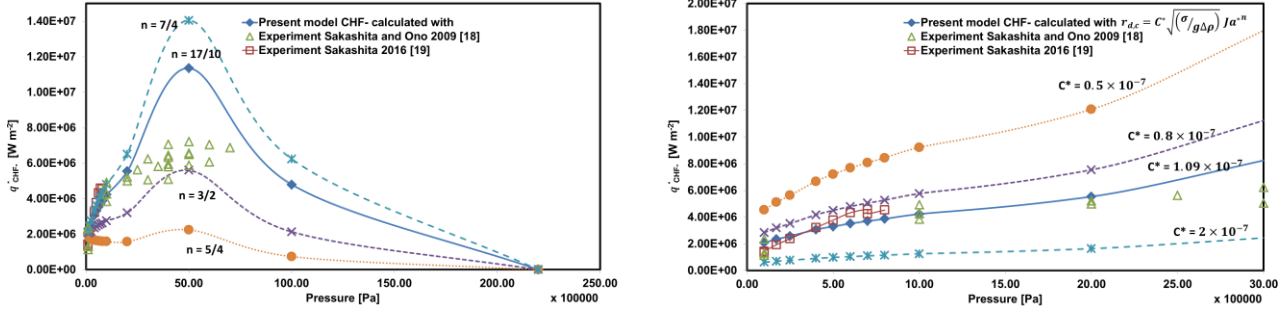
289 When Equation (22) is inserted into Mikic's bubble growth equation under thermal diffusion control,
 290 the departure diameter is given as
 291

$$r_{d,c} = \frac{B^2}{A} \sqrt{t_g^+} \quad (23)$$

292 with
 293

$$t_g^+ = C^{*2} \theta^2 \left(\sigma/g\Delta\rho \right) A^2 B^{-4} Ja^{*2n}. \quad (24)$$

294 The term $\sqrt{\left(\sigma/g\Delta\rho \right)}$ in Equation (22) is commonly called the ‘‘capillary constant of liquid’’ [43],
 295 which can be considered as a property of the liquid. We calibrate the model and compare it with the
 296 data from Sakashita [18, 19]. We directly took the onset temperature at different pressure from the
 297 experimental data [19] as a correlation $\Delta T_{sup} = -1.94 \ln(p) + 32.261$. The dependency of the liquid
 298 contact angle on the temperature is also accounted for by the expression $\theta[deg] = 157 - 0.55T[^\circ C]$
 299 [19]. Due to the small heater surface diameter in this particular vertical boiling experiment (7 mm) and
 300 because the experimental results are not obviously different from horizontal boiling ones, the impact of
 301 natural circulation to the rewetting can be neglected. The surface tension is considered as a linear
 302 function of temperature, which is $0.076 - 0.076/380[K] T_{sup}$. It is found that n determines the slope
 303 of curve and C^* determines the \dot{q}_{CHF-} at 1 bar (See *Figure 3*). The average error ($abs(exp -$
 304 $sim)/exp \cdot 100\%$) between the calculated \dot{q}_{CHF-} and experimental value was considered. The n and
 305 C^* value which leads to the lowest error will be selected.



306
 307 *Figure 3: Predicted \dot{q}_{CHF-} based on our model with “n” and “C*” introduced in Equation (22)*
 308 *compared to measured CHF by Sakashita 2009 [18] and 2016 [19] for pool boiling in water at*
 309 *different pressures.*

310 From the calibration we get $C^* = 1.09 \times 10^{-7}$ and $n = 1.7$ as values for water to calculate $r_{d,c}$. The
 311 bubble departure radius does also depend on the bulk temperature because the bulk temperature
 312 impacts the bubble growth and further the base diameter. So we calculate the departure radius basing
 313 on Equation (23) as
 314

$$r_{d,c} = C_1 C^* \left(\frac{\sigma}{g \Delta \rho} \right)^{1/2} Ja^{*n}. \quad (25)$$

315 The evaporation of the superheated liquid trapped in the cavity and the microlayer on the wall (later
 316 both are called microlayer) and liquid in thermal layer surrounding the bubbles is considered as the
 317 main contribution to bubble growth. The microlayer has a superheat of $(T_w - T_{sat})$, while the thermal
 318 layer, whose temperature decreases from T_w to T_b , has an average superheat of $\left(\frac{T_w - T_{sat}}{2} \right)$ in the
 319 superheated part and subcooling of $\left(\frac{T_{sat} - T_b}{2} \right)$ in the subcooled part when the temperature profile is
 320 assumed to be linear. From the thermal diffusion controlled bubble growth rule ($r = B\sqrt{t}$), the bubble
 321 radius should be proportional to liquid superheat ΔT_{sup} . From the Kolev’s work [42], it is found that
 322 the bubble departure diameter firstly increases and then decreases with increase of wall superheat.
 323 When the CHF is approached, the departure diameter decreases to a very small value. Firstly we
 324 assume the thermal layer thickness and bubble size is comparable when the process approaching CHF.
 325 Because the bubble growth time near the CHF is quite short, in saturated boiling the contribution from
 326 the superheated part of the thermal layer is simplified as $\frac{1}{2}$ of that from the microlayer due to the
 327 different superheat. However in subcooled boiling, the contribution from the superheated part of the
 328 thermal layer is only $\frac{(T_w - T_{sat})}{(T_w - T_b)}$ when the thermal layer is superheated in the saturated boiling case.

329 Then the contribution from the superheated part of thermal layer in this case becomes $\frac{1}{2} \frac{(T_w - T_{sat})}{(T_w - T_b)}$ of
 330 that from the microlayer. The subcooling part in the thermal layer can impact bubble growth via
 331 condensation. As it was introduced in Equation (13), the condensation rate is proportional to the
 332 subcooling temperature. C_1 can then be rewritten as
 333

$$C_1 = \frac{2}{3} \left(\frac{1}{2} \cdot \frac{(T_w - T_{sat})}{(T_w - T_b)} + 1 \right) \cdot \frac{(T_w - T_{sat})}{\left(\frac{T_{sat} - T_b}{2} \right)}. \quad (26)$$

334
 335 This holds only when the bulk temperature $\left(\frac{T_{sat} - T_b}{2} \right)$ is larger than $(T_w - T_{sat})$. Under the same
 336 conditions, X in Equation (16) can become larger than 1. This is reasonable, but still a simplification

337 and future work will be directed into that direction. In this work, the molecular effusion effect [8, 11,
 338 46] which makes the dependency of CHF on subcooling may no longer be given in the high subcooling
 339 case is not considered.

340 2.2.2 CHF- in forced convective boiling

341 We consider the flow in forced convective boiling as fully developed. The heat transfer can then be
 342 calculated with Newton's law as

$$343 \quad \dot{q} = h_c(T_w - T_b). \quad (27)$$

344 h_c is the heat transfer coefficient and may be obtained from a Dittus-Bielter type equation [47] for
 345 multiphase fluid in a pipe as

$$346 \quad h_c = 0.023 \frac{k_l}{D} \left(\frac{DG(1-\phi)}{\mu_l} \right)^{0.8} \left(\frac{\mu_l C_{p,l}}{k_l} \right)^{0.4} \cdot F \equiv \frac{\dot{q}}{(T_w - T_b)} = \frac{k_l \frac{\partial T}{\partial y} |_{y=0}}{(T_w - T_b)}. \quad (28)$$

347 D is the diameter of pipe, G is mass load, ϕ is the weight fraction of vapour and F is a function of the
 348 two-phase pressure drop which is introduced by Chen in 1966 [47].

349 Both of thermal diffusivity and forced convection can impact the thermal layer recovery in the forced
 350 convective boiling. From the calculation it is found that the impact of thermal diffusivity is much
 351 higher than that of convective flow. Then Equation (16) can be also applied for forced convective
 352 boiling. The departure radius is also impacted by shear stress on the wall τ_w , which is proportional to
 353 $\frac{\partial u}{\partial y} |_{y=0}$. The heat transfer coefficient h_c depends on τ_w and is a function of the mass load, the pipe
 354 diameter and length of the forced convective boiling area. The flow velocity profile is proportional to
 355 the temperature profile with function of $Pr^{1/3}$. Hence, the relationship between the heat transfer
 356 coefficient and the shear stress can be written as

$$357 \quad \tau_w = \mu \frac{\partial u}{\partial y} |_{y=0} = \mu \frac{h_c u_{max}}{k_l (Pr^{1/3})^{1/7}} \quad (29)$$

358 with

$$359 \quad u_{max} = \frac{120}{98} \bar{u} \quad (30)$$

360 following the turbulence 1/7 law. The surface tension force in a regular spherical bubble is given as

$$361 \quad \Delta p_s = \frac{2\sigma}{r_s}. \quad (31)$$

362 However in the flow boiling, the bubble will be inclined as a truncated spherical bubble. But we are
 363 still able to assume the surface tension force in the wall tangential direction is C_2 times of that in the
 364 regular spherical bubble. Klausner found the ratio of surface tension force in the tangential direction to
 365 that of the perpendicular direction is around 0.4 from his experiments and concluded that compared to
 366 lift off, sliding can be easier achieved with some shear stress [48]. However, this value cannot be
 367 applied here because the bubble near CHF is far different from an ordinary bubble, that is, it has much
 368 smaller diameter and shorter growth time. Sliding happens when shear stress (τ_w) is larger or equal to

369 the surface tension in the tangential direction ($C_2\Delta p_s$). Hence from Equation (29) and Equation (31),
 370 we define the sliding radius as
 371

$$r_s = \frac{C_2 \cdot 2\sigma k_l Pr^{1/21}}{\mu h_c u_{max}}. \quad (32)$$

372 Base on the concept introduced by Equation (3) and Equation (4), the \dot{q}_{CHF-} in the forced convective
 373 boiling can be written as:
 374

$$\dot{q}_{CHF-} = \frac{B^2}{2r_s} \left[\frac{4}{3} H_s - \sqrt{\left(\frac{4}{3} H_s\right)^2 - \frac{X(k_l(T_w - T_{sat}))^2 \pi}{\alpha_l B^2}} \right], \quad (33)$$

375
 376 Because r_s is dependent on void fraction (See Eq. (28) and (32)), the \dot{q}_{CHF-} in Eq. (33) is also
 377 dependent on void fraction which differs from the case of pool boiling shown in Eq. (16). However as
 378 reported by Celata and Mariani and other previous researchers [10, 49, 50], it is found that the channel
 379 orientation plays a role when the mass load is low. It means that the buoyancy impacts CHF. In our
 380 concept, it is because at low mass load, $Xr_{d,c}$ is smaller r_s and dominates the \dot{q}_{CHF-} . If we consider
 381 this, a general formula to describe \dot{q}_{CHF-} can be written as
 382

$$\dot{q}_{CHF-} = \frac{B^2}{2 \cdot \min(Xr_{d,c}, r_s)} \left[\frac{4}{3} H_s - \sqrt{\left(\frac{4}{3} H_s\right)^2 - \frac{X(k_l(T_w - T_{sat}))^2 \pi}{\alpha_l B^2}} \right], \quad (34)$$

383
 384 This particular dependency on $Xr_{d,c}$ and r_s leads to the interpretation that CHF may occur in two
 385 different ways. One is a local phenomenon when $Xr_{d,c}$ dominates \dot{q}_{CHF-} . The Equation (33) can be
 386 simplified as Equation (16) in this case. The CHF- will be based on local conditions and independent of
 387 the void fraction in the bulk. It usually occurs in pool boiling or forced convective boiling with low
 388 mass load. The other is a global phenomenon when r_s dominates \dot{q}_{CHF-} . Because r_s is determined
 389 by local shear stress, the CHF- will be impacted by the void fraction in the bulk which is dependent on
 390 the conditions of upstream. It occurs in forced convective boiling with high mass load. In the global
 391 phenomenon, r_s and CHF- are strongly dependent on the global operational conditions and cannot be
 392 calculated analytically. However, it may be numerically calculated from Computational Fluid
 393 Dynamics (CFD) simulations. Hence, implementing Equation (34) into a CFD code should be a good
 394 way to calculate \dot{q}_{CHF-} .
 395

396 2.2.3 Correlations to calculate the CHF- in forced convective boiling

397 A correlation is required to calculate the impact of void fraction onto r_s and further \dot{q}_{CHF-} in
 398 Equation (34). This correlation will be recalibrated by the experimental data from Bergles [2] (See
 399 Figure 4). In 1963, Bergles conducted systematic experiments to study the parametric dependency of
 400 CHF for the subcooled flow boiling in circular pipes with water. In Equation (32), the departure radius
 401 r_s is a function of heat transfer coefficient h_c , which is strongly dependent on the gas void fraction. At
 402 same time the constant C_2 of the surface tension force in the perpendicular and tangential direction of
 403 the wall is also unknown. First we take one measured point as a reference point, in our case $T_{b,0} =$

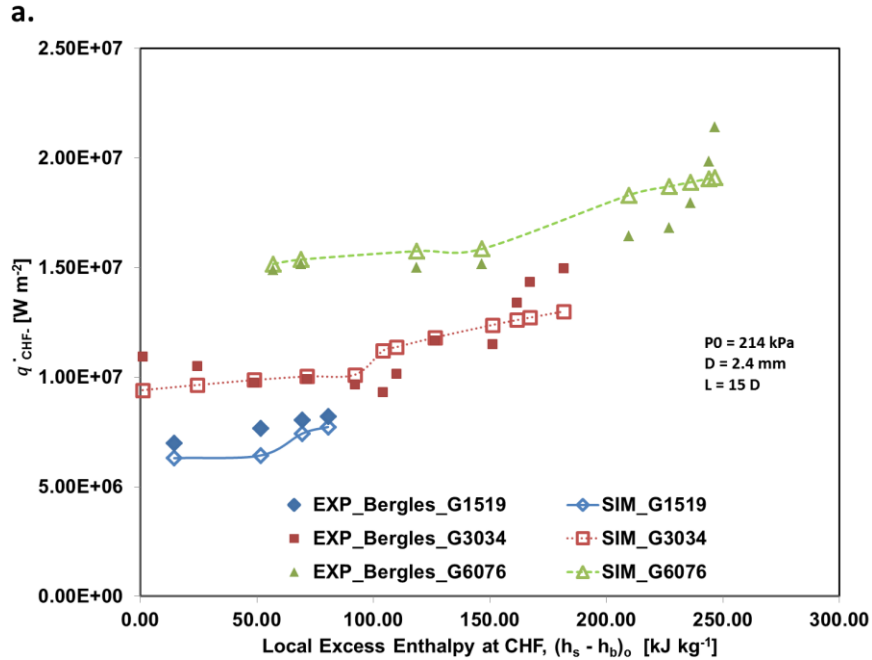
404 19 k , $G_0 = 3038 \frac{\text{kg}}{\text{m}^2 \text{s}}$, $D_0 = 1.2 \text{ mm}$, $L_0 = 60 \text{ mm}$ and $P_0 = 207 \text{ kPa}$. Based on this reference
 405 point we get $\frac{C_2}{h_c} = \frac{0.000182}{h_{c,s}}$ with $h_{c,s}$ being the heat transfer coefficient for single phase heat transfer.
 406 Then from the available 65 data points the following correlation to calculate the term $\frac{C_2}{h_c u_{max}}$ in
 407 Equation (32) is derived
 408

$$\tau_s = \frac{C_2 \cdot 2\sigma k_l Pr^{1/21}}{\mu h_c u_{max}} = \frac{0.000364 \cdot \left(\frac{D}{D_0}\right)^{0.35} \left(\frac{L}{L_0}\right)^{0.25} \left(\frac{P}{P_0}\right)^{-2.9} \left(\frac{G}{G_0}\right)^{1.15} f(\Delta T_b)}{h_{c,s} u_{max}} \cdot \frac{2\sigma k_l Pr^{1/21}}{\mu} \quad (35)$$

409
 410 Thereby $f(T_b)$ is a function of the bulk temperature T_b which is given as

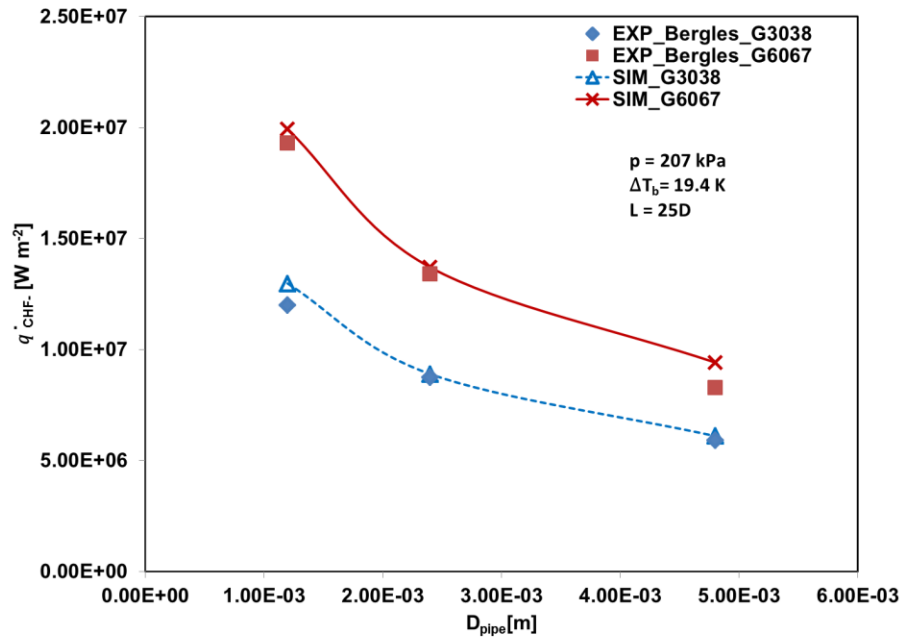
$$f(T_b) = \begin{cases} \left(\frac{26 [K]}{\Delta T_b} \cdot \frac{h_{c,s}}{h_{c,s,0}}\right)^{0.25} & \Delta T_b > 26 [K] \cdot \frac{h_{c,s}}{h_{c,s,0}} = 7.32 \times 10^{-4} [K^2 m^2 W^{-1}] h_{c,s} \\ 1 & \Delta T_b \leq 7.32 \times 10^{-4} [K^2 m^2 W^{-1}] h_{c,s} \end{cases} \quad (36)$$

411 $h_{c,s,0} = 3.55 \times 10^4 \text{ W m}^{-2} \text{ K}^{-1}$ is the single phase heat transfer coefficient at the reference point
 412 calculated with Eq. (28) with $F = 1$ and $\varphi = 0$. The calibration also based on the average error
 413 between calculated \dot{q}_{CHF} and experimental error. The calibration process and the impact of different
 414 variables in Eq. (35) and C_2 are shown in Figure 4. CHF usually happens randomly without any
 415 schematic order at the downstream part near the end or just at the end of pipe or channel [51-54]. In our
 416 case, for simplification, we consider that CHF always happens at the end of the pipe. The excess bulk
 417 temperature measured in the Bergles' case is ΔT_b in the calculation of τ_s .



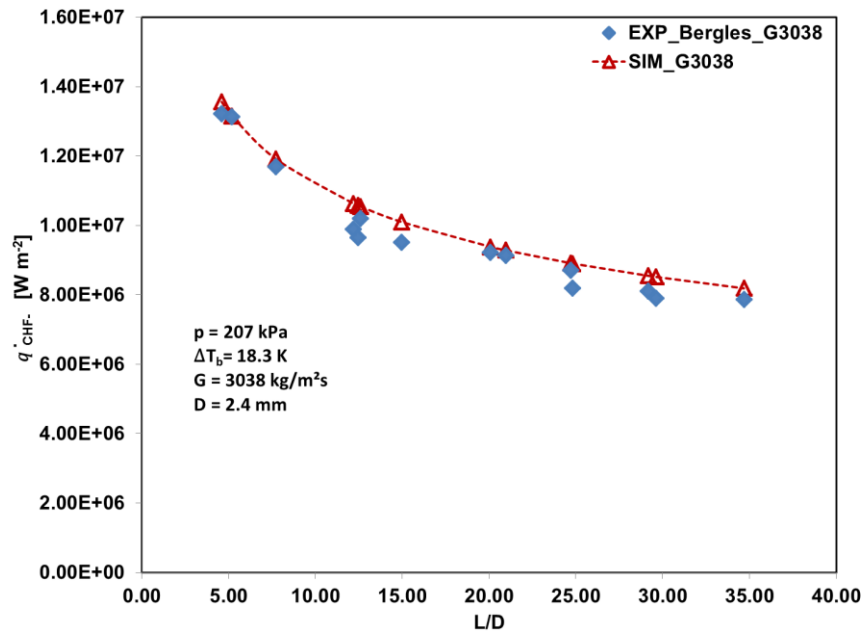
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b.

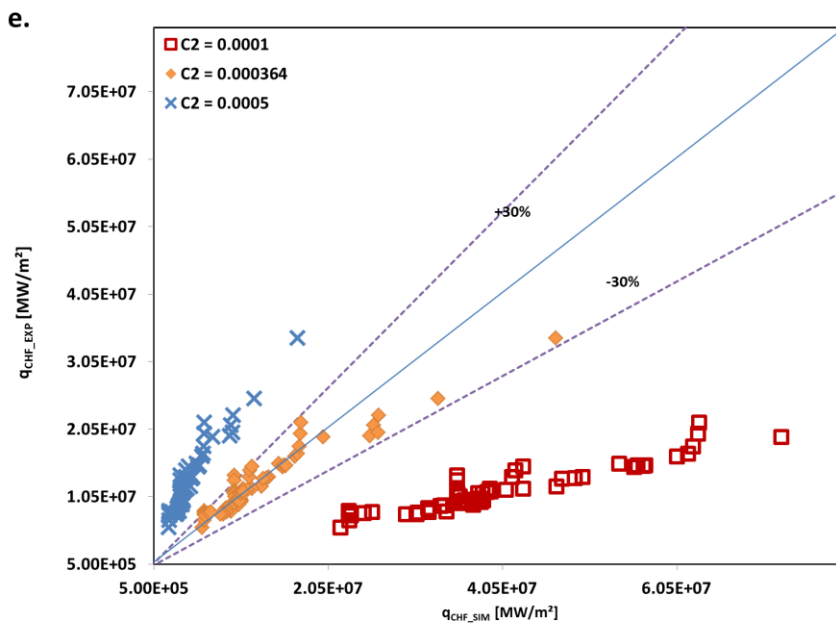
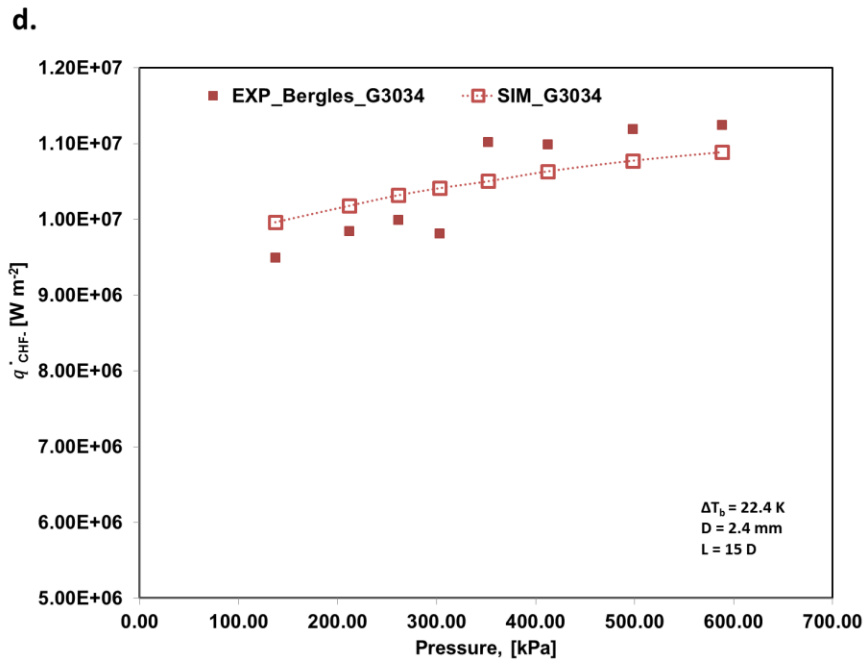


419

c.



420



421

422

423 *Figure 4: Modeled and experimentally measured \dot{q}_{CHF-} (Bergles [2]) with a) subcooling (given as*
 424 *local excess enthalpy in the plot); b) different pipe diameters (D); c) length diameter ratio (L/D), d)*
 425 *pressure (p) and e) sensitivity analysis of constant C_2 .*

426

427 The average error between the correlation and the experimental data from Bergles is around 9%. This
 428 correlation is preferred to be applied to validate the CHF calculation of other forced convective boiling
 429 cases under other operational conditions. Due to the correlations are made from the experiments of
 430 circular pipe with water, the present model is only limited for the boiling process in a circular pipe with
 431 water.

431

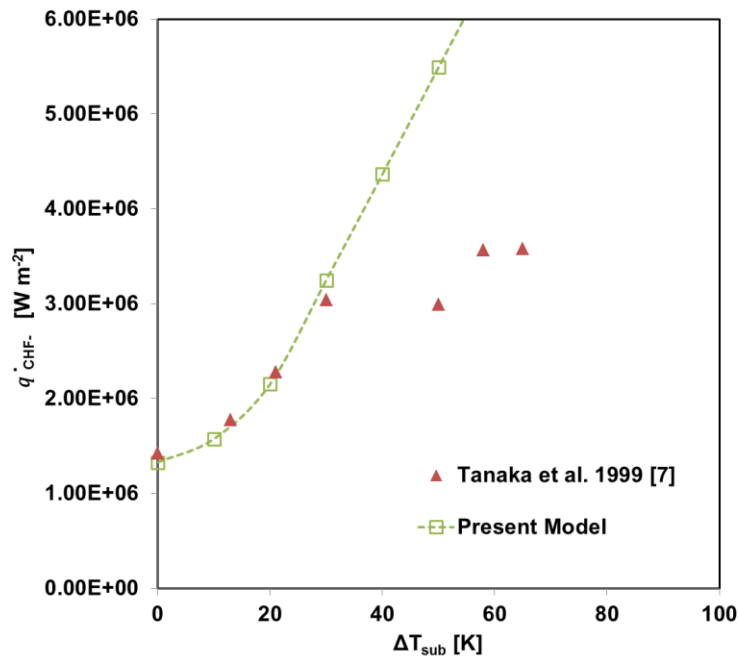
432 At present, due to the lack of information about the boiling onset, a calibration from the experimental
 433 data is needed. This calibration even can bring the impact of the wall thickness into the \dot{q}_{CHF-}
 calculation.

434 **3. Validation**

435 As is introduced, in the model there are three parameters are generated from former experiments: C^*
 436 and n are used to include the mutual effect and C_2 is used to include the bulk void fraction in the
 437 pipe. These three parameters are fixed and further applied in the validation with other experimental
 438 data.

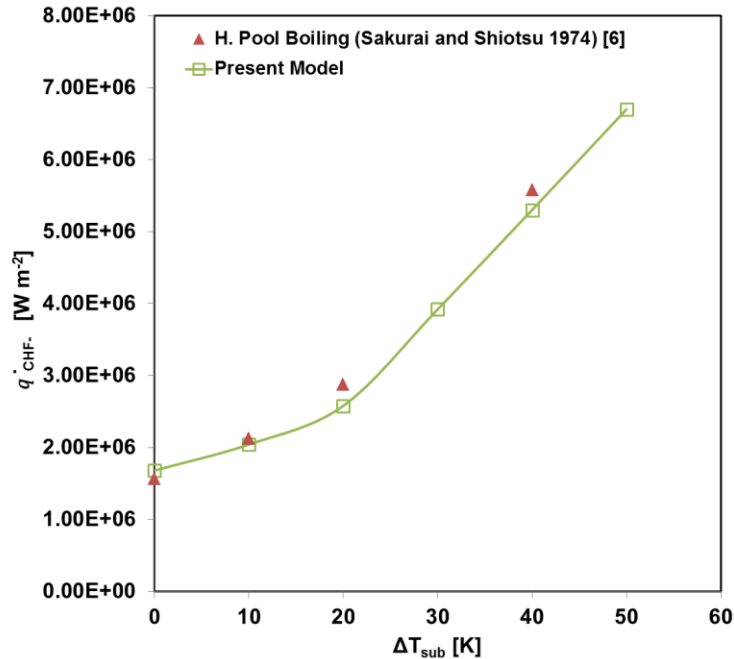
439 In 2007, Gerardi [37] performed a series of saturated pool boiling experiments with de-ionised water.
 440 He employed an IR camera and a high speed camera to observe the boiling process on an ITO heater
 441 and sapphire substrate. The measured CHF is an average value from three cases. The measured onset
 442 superheat is ~ 7 K. The average liquid contact angle of de-ionised water with the heater is 90° . With
 443 that the critical departure diameter $r_{d,c}$ is calculated as 6.25×10^{-5} m. With Equation (34) we
 444 calculate the value of $\dot{q}_{CHF-} = 1.06 \times 10^6 \text{ W m}^{-2}$ while the averaged experimental value is
 445 $9.74 \times 10^5 \text{ W m}^{-2}$. The difference is $\sim 8.8\%$ $((EXP - CAL)/EXP \times 100\%)$.

446 Also we compared our model with the experiments of Tanaka et al. [7], that is horizontal pool boiling
 447 with water under 1 atm, with different subcooling temperatures (0 – 70 K) (See *Figure 5*). In this case,
 448 the onset temperature of boiling is unknown. We firstly calibrate the onset superheat based on the CHF
 449 value in the saturated boiling case and the value is found as 7.9 K. The average error is around 31%
 450 because the molecular effusion effect is not considered in this work.



451 *Figure 5: Measured and predicted \dot{q}_{CHF-} in pool boiling as a function of subcooling when molecular*
 452 *effect is not considered (experimental data from Tanaka et al. [7] with water at 1 atm pressure).*
 453

454 In the case of Sakurai and Shiotsu [6] the authors provided a data set for horizontal pool boiling of
 455 water at one atmospheric pressure. The calibration is done for the onset wall superheat based on the
 456 case at saturated boiling. The onset wall superheat is 8.8 K. The average error for the complete
 457 calculation is $\sim 6\%$ for both horizontal case. Different to the Tanaka's case, the molecular effusion
 458 effect doesn't present in the Sakurai and Shiotsu's case (See *Figure 6*).



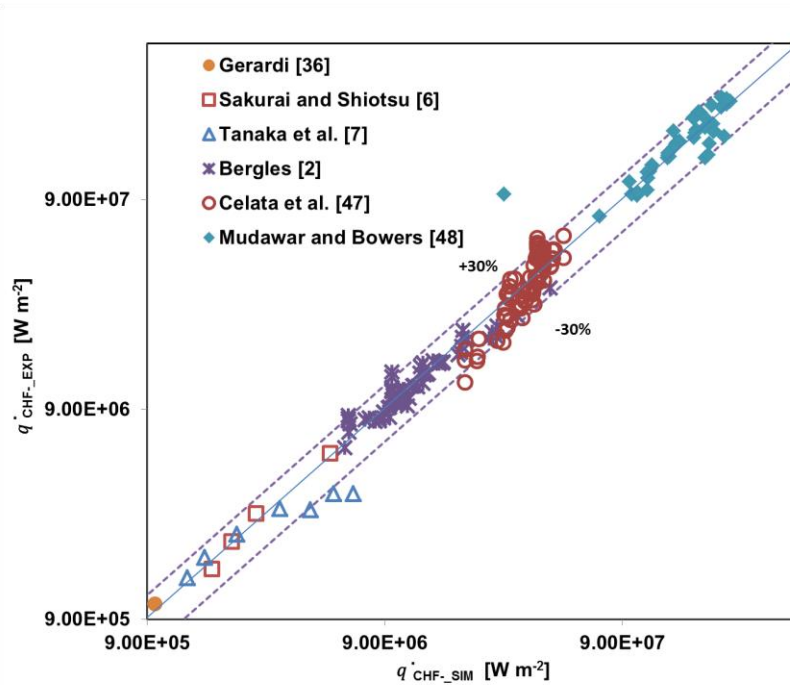
459
 460 *Figure 6: Measured and predicted \dot{q}_{CHF-} in horizontal when molecular effect is not considered*
 461 *(experimental data from Sakurai and Shiotsu [6] with water at 1 atm pressure).*

462 The CHF- model with the correlation from Bergles [2] for forced convective boiling was also validated
 463 with data from Celata [44] and Mudawar [45]. The experimental conditions are listed in the Table 1.

464 Table 1 Operational conditions of experiments for water.

	Gerardi	Tanaka et al.	Sakurai and Shiotsu	Bergles	Celata et al.	Mudawar and Bowers
Mass load (G) [kg m ⁻² s ⁻¹]	p. boiling	Pool boiling	pool boiling	1519 ~ 24276	11390 ~ 40000	20000 ~134000
Pressure (P) [kPa]	100	100	100	137 ~ 588	584.9 ~ 2614	490 ~ 4890
Subcooling (ΔT_b) [K]	0	0 ~ 70	0 ~ 40	0 ~ 58	51 ~ 149	99 ~ 211
Diameter (D) [mm]	-	-	-	1.2 ~ 4.8	2.5	0.406 ~ 0.635
Length (L) [m]	-	-	-	0.01 ~ 0.08	0.1	0.0045 ~ 0.0108
Length/Diameter	-	-	-	4.5 ~ 35	40	7.9 ~ 26.6
Calibrated Onset Superheat [K]	No calibration 7	7.9	8.8	4.45 (2.07 bar)	0.54 (20 bar)	0.24, 0.28, 0.31 (at 31 bar)
Error	8.8%	31%.	6%.	9%	18%	12%

465
 466 Totally around 200 data points from three different experiments under different operational conditions
 467 were considered (See *Figure 7*). In Mudawar's experiments, even two different wall materials and
 468 three wall thicknesses (stainless steel (SS-304) 0.406 mm and Cu-Ni 30% 0.500 mm and 0.635 mm)
 469 were involved. It was found that, the calibration of the onset point can account for the impact of the
 470 wall thickness in the calculation, which is shown in Table 1.
 471



472
473 *Figure 7: Measured and predicted \dot{q}_{CHF-} for different experiments.*

474 The average error for the case of Bergles is $\sim 9\%$, but it should be noted that the correlation was
 475 derived from these data. The average error for the case of Celata is $\sim 18\%$. The conditions of Celata
 476 and Bergles do only slightly overlap. The average error for the case of Mudawar is $\sim 12\%$, which has
 477 completely different conditions compared to that of Bergles, that is, higher pressure, higher mass load,
 478 smaller diameter and shorter length. Because the present model requires the subcooled temperature at
 479 outlet as an input condition, it is still problematic for the validations with the huge amount available
 480 published CHF experimental data from literatures.

481 **4. Conclusions and outlook**

482 A new CHF- model and its application as a model for pool boiling and flow boiling are introduced
 483 in this paper. This model is one of the very few CHF explanations, which try to explain the boiling
 484 crisis from the view point of “on the wall” instead of “near the wall” conditions. The model accounts
 485 for the impact of the different parameters, such as pressure, orientation angle, subcooling, and mass
 486 flow, hydraulic diameter, length, pressure, orientation angle in the pool or forced convective boiling
 487 and further potentially wall thickness, wettability, surface tension and so on.

488 For pool boiling CHF can be considered as a local phenomenon, while for flow boiling, there are
 489 two different phenomena leading to CHF. CHF at low flow rate is similar to that in pool boiling and
 490 considered as a local phenomenon. CHF at high flow rate is found to be a global phenomenon, which
 491 strongly depends on the upstream void fraction near wall. The criterion, which one of these phenomena
 492 is relevant, is determined by the fact, which one has the main impact on the departure diameter: shear
 493 stress (hydrodynamic) or liquid property (thermal property). The CHF- model is derived from the
 494 nucleation boiling, which allows the boiling process to continuously change from nucleation boiling to
 495 CHF-. The initiating mechanism of CHF can be explained by this model. CHF will be strongly
 496 dependent on the onset wall superheat of the cavity. In the other words, the wettability and roughness
 497 can impact the onset point and impact the CHF further. Additionally the wall thickness plays a role in
 498 the heat up of the heated liquid trapped in the cavity, which can also be considered as an impacting

499 parameter to CHF in the future. Last but not least, our model can easily be implemented in a CFD code,
 500 which would allow modelling the whole boiling process covering the nucleation boiling and boiling
 501 crisis simultaneously in one model.

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505 Nomenclature

506	A	a parameter defined as $\left(\frac{\pi h_{fg} \rho_g \Delta T_{sup}}{7 \rho_l T_{sat}}\right)^{\frac{1}{2}}$
507	\bar{A}	spatial average value of parameter A
508	B	a parameter defined as $\left(\frac{12}{\pi} \alpha_l\right)^{\frac{1}{2}} Ja$
509	c_{pl}	specific heat capacity of liquid
510	c_{pw}	specific heat capacity of wall
511	$C, C_{1,2} C^*$	constant or parameter
512	D	hydraulic diameter
513	F	a function of the two phase pressure drop factor
514	G	flow rate in a pipe
515	H	wall heat capacity per area
516	h_c	heat transfer coefficient
517	$h_{c,s}$	heat transfer coefficient for single liquid phase
518	h_{fg}	latent heat
519	Ja	Jakob number
520	Ja*	modified Jakob number $\rho_l c_{pl} T_{sat} / \rho_g h_{fg}$
521	k_l	liquid thermal conductivity
522	L	length of heated wall
523	N	nucleation site density
524	P	pressure
525	P_0	pressure in the bulk liquid
526	P_s	Laplace pressure of the gas liquid interface
527	Pr	Prandtl number
528	\dot{q}	heat flux
529	\dot{q}_{CHF-}	critical heat flux
530	Q	heat of bubble life cycle
531	Q_b	heat in the bubble
532	$Q_{b,c}$	heat due to condensation around the bubble
533	$Q_{b,s}$	heat flowing from superheated liquid surrounding the bubble
534	$Q_{b,w}$	heat of bubble from wall
535	$Q_{l,c}$	heat required to reheat the trapped liquid in the cavity
536	Q_q	quenching heat
537	Q_r	residual heat in the wall around the cavity
538	Q_t	latent heat of the evaporated liquid in the cavity

539	r	radius
540	r^+	dimensionless radius
541	r_c	critical nucleus radius
542	r_s	sliding radius due to shear stress
543	r_d	bubble departure radius
544	$r_{d,c}$	minimum bubble departure radius
545	T_b	bulk temperature
546	T_l	liquid temperature
547	T_w	wall temperature
548	T_{sat}	saturated temperature
549	T_{sub}	sub cooling temperature
550	ΔT_{sup}	superheat
551	$\overline{\Delta T_{sup}}$	average superheat of the wall
552	ΔT_b	subcooling
553	t	time
554	t^+	dimensionless time
555	t_a	cavity activation time
556	t_c	bubble condensation time
557	t_g	bubble growth time
558	t_r	thermal layer recovery time
559	t_w	bubble waiting time
560	τ_w	shear stress
561	τ_g	average bubble growth time
562	u_{max}	maximum velocity in the pipe
563	\bar{u}	average velocity
564	\bar{V}	Average fluctuation velocity
565	X	a parameter defined as $1 + \frac{(T_w - T_{sat})^2}{(T_b - T_{sat})^2}$
566	α_l	liquid thermal diffusivity
567	β	orientation angle
568	φ	weight fraction of the vapor
569	θ	liquid solid contact angle
570	σ	surface tension
571	μ	viscosity
572	ρ_g	density of vapor
573	ρ_l	density of liquid
574	ρ_w	density of wall
575	Δp_s	surface tension in perpendicular direction

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