Flood-cooling of Mechanical Components

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Abstract: A methodology is presented for evaluating the safety of an accidental or intentionally applied accelerated cooling process. Accelerated cooling occurs due to increased heat transfer from the wetted surfaces. For steady state processes like pool boiling, the change across the various heat transfer regimes occurs at specific temperatures and due to extensive studies, the minimum and maximum limits of heat flux can be determined with reasonable accuracy. However, flood-cooling is not a steady state process and the boiling curve exhibits some transient distortion. A flood-cooling heat transfer model was developed based on a generalized correlation of single and two-phase forced flow combined with literature data for the modified heat transfer regimes during quenching of hot surfaces. The model was implemented as an ABAQUS user subroutine with the flood-boiling heat transfer coefficient depending on the surface geometry, orientation, thermo-physical properties and temperature, as well as water temperature and flow rate. The suggested model was validated by simulating quenching and bottom flooding experiments found in the literature. The methodology and user subroutine can be used in coupled temperature-displacement or sequential heat transfer-stress analyses of machine parts being flood cooled to iteratively determine the range of process parameters (water flow rate and water temperature) that would ensure that the thermal stresses during internal flood-cooling do not exceed critical values.

Keywords: Accelerated Cooling, Boiling Heat Transfer, Bottom Flooding, Flood-cooling, Flow Boiling, Heat Transfer, Pool Boiling, Process Optimization, Quenching, Thermal Stresses, Water Cooling.

1. Introduction

This paper discusses a methodology developed to accurately simulate and optimize the process of flood cooling with water. Accelerated cooling is sometimes used when machinery is taken off-line for regular maintenance or in the case of an emergency shut-down so that the temperature is lowered to levels that allow direct access as soon as possible.

Section 2 describes the heat transfer phenomena taking place during water flooding. A user subroutine based on flow boiling heat transfer correlations was developed in order to calculate the transient heat transfer coefficient on the wetted surfaces for finite element simulations. The validity of the selected correlations for flood cooling applications was determined by simulating an experiment described in the literature. A discussion of the finite element model details, user subroutine implementation as well as a comparison of the experimental and calculated cooling curves is provided in Section 3. Section 4 summarizes the conclusions and the proposed application of the methodology in optimization of the flood cooling process.

2. Heat transfer during flood cooling

Depending on the temperature of the mechanical part that is being subjected to flood cooling, the heat transfer mode can be single phase convection or, if the temperature is high enough for boiling to occur, two-phase convection. If the temperature of the part is even higher, a continuous film of vapor may cover the flooded surface limiting heat transfer to conduction and radiation across the film thickness.

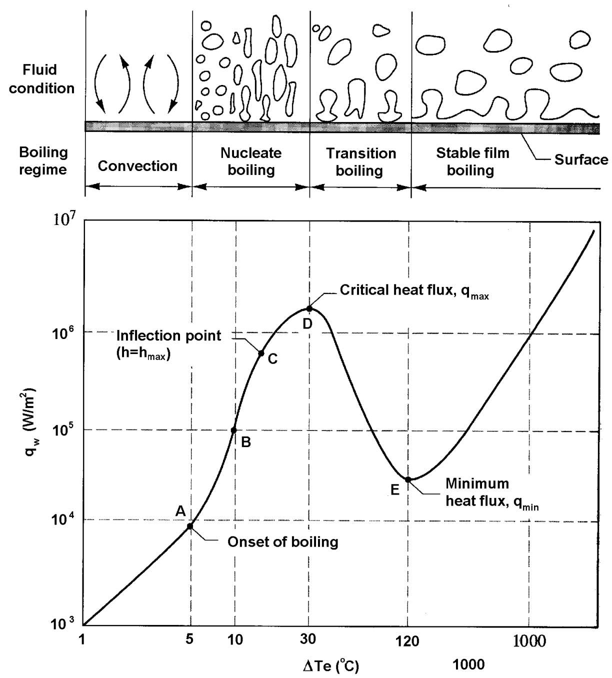

Figure 1. Heat transfer regimes and typical boiling curve for water at atmospheric pressure (Tong, 1997).

Even though the subject of boiling heat transfer has been extensively studied in the past, the phenomena taking place across the various boiling regimes are still not completely understood. Boiling is the phenomenon associated with phase change from liquid to vapor having as a consequence increased heat transfer (Bejan, 2004). Pool boiling occurs on a hot surface submerged in a pool of still liquid. Flow boiling involves a moving stream of liquid or a mixture of liquid and vapor (Cengel 2003). Quenching and flood-cooling also involve flow, at least during the initial stages of the process. If the surface temperature is higher than the saturation temperature, at which boiling starts at a given pressure, the difference between the two is called the wall superheat (Hagen, 1999). Boiling heat transfer, as seen in Figure 1 as a log-log plot of heat flux versus wall superheat, is marked by four different regimes: convection, nucleate boiling, transition boiling and film boiling. These regimes are similar in both pool and flow boiling, with buoyancy being dominant in pool boiling and the combination of both fluid motion and buoyancy driving the process in flow boiling. A higher flow velocity generally increases heat transfer in the transition and film boiling region.

In order to realistically model heat transfer during the flood cooling process, a series of appropriate correlations was selected from the literature. These correlations were developed based on, whenever possible, a physical basis but more often based on a large number of experimental data. The formulas are given in the Appendix and were implemented in the form of a User Subroutine in order to allow generation of the transient boiling curve and derivation of an instantaneous local heat transfer coefficient on the flooded surfaces.

3. Validation

To qualify the validity of the selected heat transfer correlations for the flood cooling application, the developed user subroutine was used to simulate the transient heat transfer during quenching of a vertical hot surface with bottom flooding according to the description of an experiment found in the literature (Mitsutake, 2003) The calculated transient temperatures were in good agreement with the cooling curves recorded during the experiment.

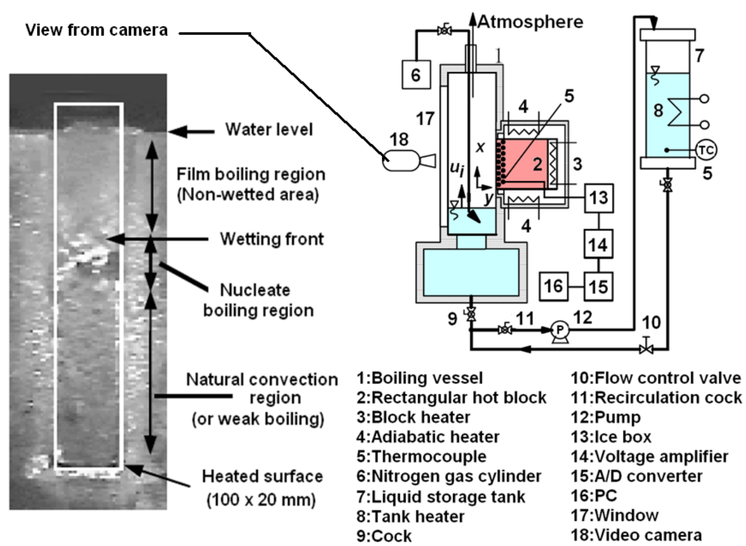

Figure 2. Surface boiling snapshot and experimental apparatus (Mitsutake, 2003).

Figure 2 shows the experimental setup (Mitsutake, 2003) consisting of three main parts with several items each; a) a boiling vessel, b) a liquid supply system and c) a data acquisition system. A 100 mm-high, 20 mm-wide, 120 mm-deep copper block shown in Figure 2 as item (2) is heated uniformly at 300 ^oC before 20 ^oC water is allowed to bottom-flood the 100 mm-high, 20 mm-wide front surface at a rate of u_i=1.29 mm/s.

A transient heat transfer analysis was run for a total of 150 s. The finite element model was a 100 mm-high, 10 mm-wide, 120 mm-deep rectangular block taking into account the plane of symmetry at z=0 with a mesh of about 7,250 quadratic hexahedral elements of type DC3D20 (Dassault Systèmes, 2009) as shown in Figure 3. The block was assumed to be copper with a density of 8933 kg/m^3, a specific heat of 400 J/kg K and a thermal conductivity of 380 W/mK. An adiabatic boundary condition was also assumed for the external surfaces, except for the cooled side. The cooled side corresponds to the Y=0 plane in Figure 3. The local heat transfer coefficient on that side was calculated by the user subroutine. Coding in the subroutine ensured that all points above the flood plane had a heat transfer coefficient of 5.3 W/m^{2}K, appropriate for natural convection whereas heat transfer coefficients for all the points below the flood plane depended primarily on water temperature, flow velocity and surface material and were calculated based on the correlations given in the Appendix.

During the experiment, temperature histories were recorded at various locations within the block. Figure 3 shows the calculated temperature field 17 s after the start of bottom flooding.

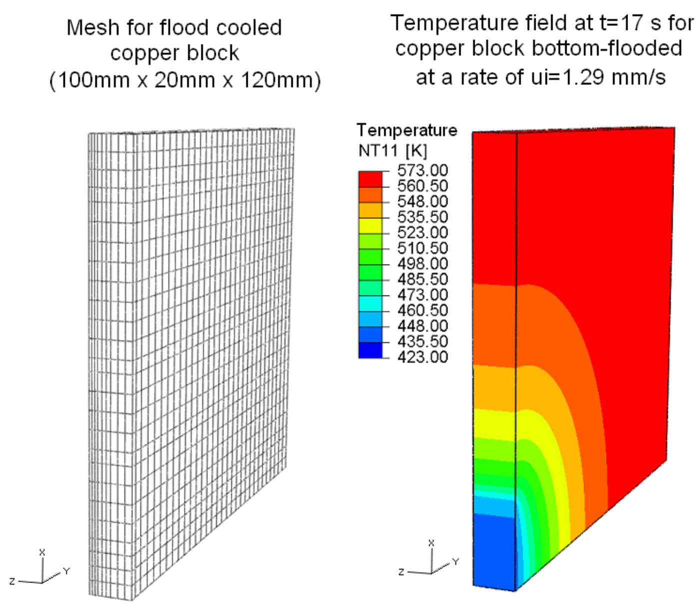

Figure 3. Finite element mesh and temperature field 17 s after the start of flooding.

Figure 4 shows a comparison of the experimentally obtained temperature histories and the calculated transient temperatures at the same locations. The coordinate system referenced by the X, Y and Z values is shown in Figure 3.

Generally, a good agreement is observed between the experimental and the calculated cooling curves shown in Figure 4, given the fact that different correlations were derived from different subsets of experimental data. According to information found in the literature, (Kandlikar, 1990), the calculated estimates for the heat transfer coefficients can deviate by 12%-20%, when compared to a broader set of actual data.

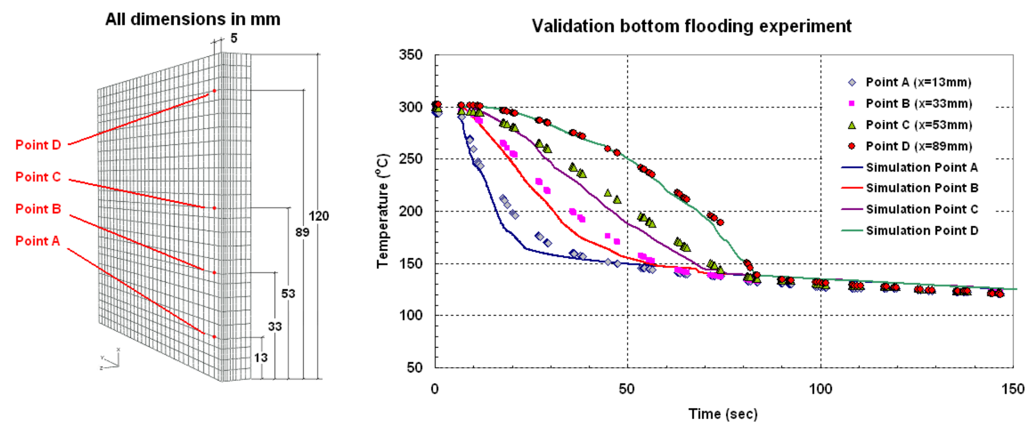

Figure 4. Comparison of experimental and calculated temperatures at points (X=as shown, Y=5 mm, Z=0).

4. Conclusions and flood cooling process optimization

This paper has discussed a heat transfer correlation-dependent user subroutine developed to calculate local heat transfer coefficients on a surface flooded with water. The validity of the selected correlations was confirmed by simulating a vertical wall being cooled by bottom flooding and comparing the calculated transient temperatures with cooling curves at approximately the same locations given in the literature.

The developed subroutine can be used to optimize the process of flood cooling with water if needed for machinery that is normally operated at high temperature and occasionally needs to be stopped and cooled down to temperatures that allow physical access for inspection, cleaning or maintenance. The steep temperature gradients occurring during the cooling process, however, can result to thermal stresses above critical levels. A coupled temperature displacement analysis taking advantage of the developed subroutine can enable iterative identification of the water flow velocity and water temperature that can lead to faster heat removal while ensuring that the thermal stresses are acceptable.

5. References

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6. Acknowledgement

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7. Appendix: Boiling heat transfer correlations

When boiling occurs under transient conditions, the “classic” steady state boiling curve shown in Figure 1 will get distorted but the basic boiling regimes of convection, nucleate boiling, transition and film boiling will remain the same.

Depending on the hot surface and the water temperatures as well as the type of solid material, surface roughness and flow conditions, the onset of nucleate boiling with a consistent flow of bubbles occurs at temperature T_{ONB} which is above the saturation temperature, T_{sat}. If the surface is at temperatures below T_{ONB}, the heat transfer mode is single-phase convection.

Above T_{ONB} heat transfer occurs by a combination of nucleate boiling and single-phase convection. The relative contribution of single-phase convection and nucleate boiling can be estimated depending on the solid/water temperatures and flow conditions.

Depending on the water temperature, nucleate boiling can be saturated or subcooled. Saturated boiling occurs if the water used for cooling is above its saturation temperature T_{sat}. If the water temperature is below the saturation temperature T_{sat}, boiling is called subcooled.

Subcooled flow boiling encompasses a partial boiling regime where single-phase convection is significant, a fully developed boiling regime where nucleate boiling is essentially the main mode of heat transfer, and, if the surface temperatures are high enough, there can be a point where the void (vapor) fraction begins to be significant and heat transfer is better estimated by the corresponding saturated flow boiling correlation.

Subsequent paragraphs describe the methods applied to estimate the surface temperature T_{FDB} for which transition from partial to fully developed boiling occurs, as well as the temperature, T_{NVG}, where the net vapor generation is high enough to have a significant void flow boiling regime.

Nucleate boiling is stable until a critical heat flux is reached, at which point boiling heat transfer enters the transition regime. Correlations for the surface temperature, T_{CHF}, associated with the critical heat flux for various combinations of solid materials and cooling fluids with various degrees of subcooling can be found in the literature.

The heat transfer phenomena in the transition regime are very complex and few correlations are available. Transition boiling will occur if the surface temperature is above T_{CHF} and below the minimum stable film boiling temperature T_{MFB}. This temperature depends on the degree of subcooling, the combination of solid material and fluid coolant, the surface condition and the flow parameters.

Finally, if the surface temperature is above T_{MFB}, a stable film will form allowing heat transfer to take place in the form of conduction and radiation through vapor.

7.1 Single-phase convection

The Dittus-Boelter (Dittus, 1930) is the most widely used empirical correlation for the heat transfer coefficient h_L from a hot surface at temperature T_w to a cold liquid at temperature T_L by single phase convection based on the Reynolds and Prandtl numbers:

$$h_L = 0.023 \cdot \mathrm{Re}_f^{\;0.8} \cdot \mathrm{Pr}_f^{\;0.4} \cdot \frac{k_f}{L_c} \tag{1}$$

where k is the thermal conductivity and L_c is the characteristic length of the flow. The subscript f corresponds to liquid properties at the average temperature:

$$T_f = \frac{T_w + T_L}{2} \tag{2}$$

7.2 Onset of nucleate boiling

Appropriate equations for the estimation of the lowest temperature T_{ONB} of the surface being cooled by water, required to initiate boiling were presented by Hsu (Hsu, 1962) and Sato and Matsumura (Sato, 1963):

$$T_{ONB} = T_{sat} + \frac{4 \cdot \sigma \cdot T_{sat} \cdot h_L}{k_L \cdot h_{fg} \cdot \rho_{fg}} \cdot \left[1 + \sqrt{1 + \frac{k_L \cdot h_{fg} \cdot \rho_{fg} \cdot \Delta T_{sub}}{2 \cdot \sigma \cdot T_{sat} \cdot h_L}} \right] \tag{3}$$

where σ is the surface tension between the solid and water, h_{fg} is the latent heat of vaporization, ρ_{fg} is the density of water at saturation temperature and ΔT_{sub} is the degree of subcooling equal to:

$$\Delta T_{sub} = T_{sat} - T_L \tag{4}$$

The subscript L corresponds to liquid properties at bulk temperature.

7.3 Saturated flow boiling

A correlation for the saturated flow boiling heat transfer coefficient was initially proposed by Kandlikar (Kandlikar, 1983) and later modified by the same author (Kandlikar,1990) so that it can be used for both horizontal and vertical flows:

$$h_{SATB} = h_L \cdot C_1 \cdot Co^{C_2} \cdot (25 \cdot Fr_L)^{C_5} + h_L \cdot C_3 \cdot Bo^{C_4} \cdot F_{fl} \tag{5}$$

Equation 5 depends on the single-phase heat transfer coefficient h_L, the Froude number with all flow as liquid, equal to:

$$Fr_L = \frac{u^2}{g \cdot L_c} \tag{6}$$

and the boiling number:

$$Bo = \frac{\dot{q}''}{u \cdot \rho_L \cdot h_{fg}}. \tag{7}$$

The parameter F_{fl} is associated with the type of fluid (equal to 1.0 for water), the water flow velocity is u, and the heat flux which is equal to:

$$\dot{q}'' = h_{SATB} \cdot \left(T_w - T_{sat}\right).$$

(8)

Another parameter involved in Equation 5 is the convection number Co which is equal to:

$$Co = \left(\frac{1-\chi}{\chi}\right)^{0.8} \cdot \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \tag{9}$$

with the vapor quality equal to: $\chi = -c_{pL} \cdot \dfrac{\left(T_{sat} - T_L\right)}{h_{fg}}$ (10)

In Equation (9), ρ_V is the density of vapor and in Equation (10), c_{pL} is the specific heat of water. The values of the remaining coefficients depend on the convection number Co and are given in Table 1.

Table 1. Values of coefficients for saturated flow boiling in the convective and nucleate boiling regions (Kandlikar, 1990)

Constant	Convective region $Co \le 0.65$	Nucleate boiling region $Co > 0.65$
C_1	1.1360	0.6683
C_2	-0.9	-0.2
C_3	667.2	1058.0
C_4	0.7	0.7
C_5*	0.3	0.3
* C_5 =0 for vertical flow and for horizontal flow with $Fr_L > 0.04$.		

7.4 Subcooled flow boiling

7.4.1 Fully developed boiling

Kandlikar (Kandlikar, 1991), developed a correlation for the heat flux in the fully developed subcooled nucleate boiling region based on the surface superheat:

$$\Delta T_{sat} = T_w - T_{sat} \tag{11}$$

This correlation states that the heat flux is equal to:

$$\dot{q}''_{FD} = \left[1058 \cdot \frac{F_{fl} \cdot h_L \cdot \Delta T_{sat}}{\left(u \cdot \rho_L \cdot h_{fg}\right)^{0.7}}\right]^{1/0.3} \tag{12}$$

7.4.2 Partially developed boiling

Bowring (Bowring, 1962) suggested a methodology to calculate the transition from partial to fully developed boiling. Based on this methodology, the heat flux at the point of transition to fully developed nucleate boiling is:

$$\dot{q}_{FDB}'' = 1.4\,\dot{q}_{\mathrm{int}}'' \tag{13}$$

with $\dot{q}_{\mathrm{int}}''$ being the heat flux at the intercept of the curves representing single-phase convection and fully developed nucleate boiling, corresponding to point F in Figure 5.

Kandlikar (Kandlikar, 1991), proposed a procedure to calculate the heat flux in the partial boiling region based on the temperatures and heat fluxes at the bounding points of the region. These points are shown as C and E in Figure 5 and can be calculated from the correlations for single-phase convection at the onset of nucleate boiling T_C=T_{ONB} and the Bowring methodology for the point of transition from partial to fully developed boiling T_E=T_{FDB}. The heat flux in the partial boiling region also depends on an exponent *m* which varies linearly from 1 to 1/0.3 between points C and E as follows:

$$\dot{q}_{NB1}'' = a + b\cdot\left(T_w - T_{sat}\right)^m \tag{14}$$

The slope and constant parameters are given as: $b = \dfrac{\dot{q}_E'' - \dot{q}_C''}{\left(T_E - T_{sat}\right)^m - \left(T_C - T_{sat}\right)^m}$, (15)

and $a = \dot{q}_C'' - b\cdot\left(T_C - T_{sat}\right)^m$. (16)

7.4.3 Significant void flow boiling

The correlations by Saha and Zuber (Saha, 1974) are used to calculate the thermodynamic quality χ_{NVG} at the transition from fully developed to significant void flow boiling that corresponds to surface temperature T_{NVG}:

For $\mathrm{Re}_L\cdot\mathrm{Pr}_L$ <70,000: $\chi_{NVG} = -0.0022\cdot Bo\cdot \mathrm{Re}_L\cdot\mathrm{Pr}_L$ (17)

For $\mathrm{Re}_L\cdot\mathrm{Pr}_L \geq$ 70,000: $\chi_{NVG} = -154\cdot Bo$ (18)

After χ_{NVG} has been determined, an apparent quality can be calculated as:

$$\chi_a = \chi - \chi_{NVG}\cdot\exp\left(\frac{\chi}{\chi_{NVG}} - 1\right)\Bigg/ 1 - \chi_{NVG}\cdot\exp\left(\frac{\chi}{\chi_{NVG}} - 1\right) \tag{19}$$

It should be noted that the parameter χ in Equation 19 is the actual vapor quality which is negative in the subcooled region. Kandlikar (Kandlikar, 1997), postulated that the heat flux in the significant void flow region can be calculated based on the nucleate boiling region $Co > 0.65$ correlations for saturated flow boiling using the apparent vapor quality.

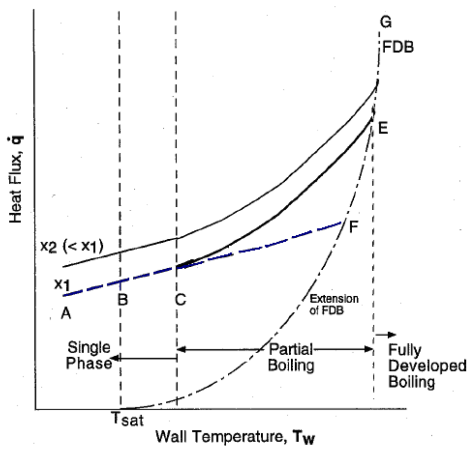

Figure 5. Partial and fully developed subcooled flow boiling. (Kandlikar, 1997)

7.5 Critical heat flux temperature

An extensive review of the existing correlations was published by Carbajo (Carbajo 1985). A recent, empirical correlation taking into account the degree of subcooling and the type of surface material and cooling fluid is expressed by:

$$T_{CHF} = T_{sat} + b_1 + a_1 \cdot (T_{sat} - T_L) \tag{20}$$

The constant b_1 is equal to 62 for aluminum, 30 for copper and 29 for stainless steel, if the temperatures are expressed in K or ^{o}C. The parameter a_1 is related to the thermal properties of the solid surface (subscript S) and the fluid (subscript L) as follows:

$$a_1 = 0.096 \cdot \frac{k_S \cdot \rho_S \cdot c_{pS}{}^2}{k_L \cdot \rho_L \cdot c_{pL}{}^2} \tag{21}$$

7.6 Minimum film boiling temperature

Carbajo (Carbajo 1985), also reviewed the various correlations for the minimum film boiling temperature and proposed a general relationship that accounts for subcooling, fluid and solid material properties, surface conditions and flow rate. For small flow rates, $u \cdot \rho_L$ <2 kg m^2/s the flow rate should be assumed to be zero.

$$T_{MFB} = T_{sat} + a_2 \cdot (T_{sat} - T_L) + \left\{ \Delta T_{MFB,ISO} \cdot (1 + \beta_2 \gamma_2) \cdot \left[1 + 0.1 \cdot (u \cdot \rho_L)^{0.4} \right] \right\} \tag{22}$$

When the cooling fluid is water and the temperatures are expressed in K or ^{o}C, $\Delta T_{MFB,ISO}$ is equal to 60 K. The remaining parameters depend on the combination of solid and fluid and representative values are given in Carbajo, (Carbajo 1985). For example, the values for water on copper are a_2 =6.13 and $\beta_2\gamma_2$ =0.167 and for water on stainless steel a_2 =6.78 and $\beta_2\gamma_2$ =2.167.

7.7 Transition boiling

Due to the limited understanding of the physical phenomena in the transition region and the difficulty in obtaining reliable data in that regime, there are no widely accepted correlations in the literature.

In order to generalize the proposed methodology, many options should be investigated and a library of functions should be included in the user subroutine. The sensitivity of the results for each specific application should be considered when the appropriate function is selected.

As an example, for quenching applications the heat transfer coefficient in the transition regime, h_{TRANS}, has been proposed (Barnea, 1994) to decrease exponentially from the value at critical heat flux h_{CHF}, according to the function:

$$h_{TRANS} = h_{CHF} \cdot \exp\left(\frac{T_{CHF} - T_w}{T_{CHF} - T_{sat}}\right) \tag{23}$$

For the purposes of this work, the simplest approach was selected and the heat flux throughout the transition regime was conservatively assumed to be equal to the critical heat flux.

7.8 Film boiling

A correlation for the heat transfer coefficient in flow film boiling is given by Collier (Collier, 1972) as a combination of a convective and a radiative component:

$$h_{film} = h_{CF} + a_3 \cdot h_R \tag{24}$$

The radiative component is equal to:

$$h_R = \frac{5.67\times10^{-8} \cdot \left(T_w{}^4 - T_{sat}{}^4\right)}{\left(T_w - T_{sat}\right)} \tag{25}$$

with all temperatures in degrees K.

The multiplication factor is a_3 =0.75 if $u < (g \cdot L_c)^2$ and a_3 =0.875 otherwise. The convective component is equal to:

$$h_{CF} = 0.67 \cdot \left[\frac{k_G \cdot \rho_G \cdot (\rho_L - \rho_G) \cdot g \cdot h'_{fg}}{\mu_G \cdot L_c \cdot (T_w - T_{sat})}\right]^{0.25}. \tag{26}$$

The augmented heat of vaporization is: $h'_{fg} = h_{fg} + 0.4 \cdot c_{pG} \cdot (T_w - T_{sat})$. (27)

Subscript (L) corresponds to water at bulk temperature and subscript (G) corresponds to vapor at saturation temperature.