Incipience of the intensive heat transfer regime at cooling high-temperature bodies in subcooled liquid

P K Kanin, V V Yagov, A R Zabirov and M A Lexin
Department of Thermophysics, NRU “Moscow Power Engineering Institute”,
14 Krasnokazarmennaya Street, Moscow 111250, Russia
E-mail: kaninpk@mpei.ru

Abstract. Cooling in film boiling is usually an unwanted process in many technologies due to low intensity of heat transfer. Thus, predicting the solid wall superheat at vapor film destabilization is useful to avoid this phenomenon. In the present paper, two new semi-empirical models for evaluation of the wall superheat at destabilization of vapor film around the metallic body cooled in saturated or in subcooled liquid are proposed. Both models with fitted empirical multipliers are in good agreement with an experimental dataset. To evaluate the contribution of the natural convection in the model for temperature head at cooling in subcooled liquid, a problem about the natural convection near the vapor film, occurring during film boiling along the vertical plane, is numerically solved by means of ANES20XE CFD-code. The computational results of longitudinal velocity are in good agreement with the theoretical velocity of natural convection used in the model.

1. Introduction
Intensive cooling of high-temperature bodies in subcooled or saturated liquid is a problem of current interest for many industries and technologies. For instance, an increase in cooling rate at quenching provides material hardening [1], and intensive heat removal at cooling of the internals of the reactor pressure vessel is important for the management and mitigation of severe accidents on a nuclear power plant with light water reactors.

During film boiling of water subcooled over 22 K on superheated surface of a metallic body, the vapor film can break-up, and the intensive heat transfer regime of "microbubble" boiling arises along with a significant increase in HTC. This regime was first mentioned in papers [2, 3] and named according to the visual observation of many tiny bubbles that appeared around the cooled copper sphere. To reveal and define the regularities of the incipience of this process, theoretical and experimental research is being performed in the Department of Thermophysics of NRU "MPEI" under the direction of professor Yagov V.V.

At high superheat of a cooled body relatively saturation temperature of the coolant, direct contact between the liquid and the cooled surface is excluded due to vapor film formation around the body. This film acts as an insulator and significantly decreases heat transfer intensity. In this case, prediction of the solid wall superheat corresponding to vapor film collapse and heat transfer increase due to transition to the more intensive heat transfer regime becomes important from a practical point of view.

A wide base of accumulated experimental data allowed us to develop a semi-empirical model of the incipience of the intensive heat transfer regime at film boiling of subcooled liquid [4]. The derivation of the model was performed by means of the method of physical estimations using the...
results of Kapitsa P. L. [5]. To date, this model has been corrected [6], and better founded equations for temperature head at cooling in both subcooled and saturated liquid have been derived.

2. Experimental and theoretical results

2.1. Experimental equipment
To study regularities of film boiling of subcooled liquid a wide experimental dataset has been accumulated. This dataset is based on experiments on cooling metallic spheres and cylinders in different liquids and covers a range of liquid temperatures from −80 to 150°C and pressures from 0.1 to 1.0 MPa.

The experimental facilities and test samples used in our studies are described in detail in [7, 8, 9]. The test samples (spheres with the diameter of 30-50 mm and cylinders of 10×45–50 mm) have up to five internally embedded K-type thermocouples with the measurement error of ±1 K.

To evaluate heat transfer coefficients an inverse heat conduction problem is being solved. Our approach to solving this problem is described in [7, 10].

2.2. Typical experimental results
A thermogram (temperature variation with time) is a result of an experiment. There are thermograms for two experiments depicted in figure 1: cooling of a 40 mm stainless steel sphere in isopropanol (a) at atmospheric pressure and liquid temperature of 62°C, and cooling of the same sample in water (b) at atmospheric pressure and liquid temperature of 80°C. This sample has four internally embedded thermocouples. There are three of them near the surface at polar angles of 90, 135 and 180°, and one in the center.

![Thermograms of cooling stainless steel sphere in isopropanol (a) and in water (b) at ΔT_{sub} = 20 K.](image)

As it is shown in our previous papers [4, 7, 8, 10, 11], the duration of cooling in stable film boiling mode depends on many factors including liquid subcooling and thermophysical properties of both liquid and material of the cooled body. It can be seen from figure 1, that at fixed subcooling and same material of the sample, the duration of cooling in isopropanol and in water in film boiling mode differs 4–5 times.

The difference appears also in the transition temperatures. The transition temperature may be defined as the average temperature of the cooled surface at destabilization of the vapor film. The moment of the vapor film destabilization may be evaluated based on visual analysis of experimental thermograms as an inflection point of the thermogram before a sharp increase in cooling rate. For the thermograms depicted in figure 1, the transition temperatures are equal to 193°C and 590°C, respectively, for isopropanol (a) and water (b). According to our experience, the transition temperature uncertainty can reach ±40 K depending on the speed of the vapor film collapse. The scatter of the
transition temperature even at similar experimental conditions is caused by strong inhomogeneity of thermophysical properties, possible changes in the microstructure of the cooled surface during repeated heating and cooling, as well as the presence of eddies in the liquid.

2.3. Theoretical results

The heat transfer intensity at stable film boiling of subcooled liquid can be evaluated using equation (1) proposed in papers [11, 12]. The reference temperature of liquid properties for this equation is the saturation temperature, and the reference temperature of vapor properties is \((T_w + T_s)/2\).

The comparison of the calculation results of equation (1) and those obtained through inverse heat conduction problem solving is depicted for the mentioned experiments on cooling 40 mm stainless steel sphere in isopropanol (a) and in water (b) in figure 2. As it can be seen, the experimental points differ from the curve predicted by equation (1) by not more than 30%.

\[
\alpha = 0.59 \left( \frac{\lambda \Delta T_{sub}}{\nu g} \right)^{1/4} \left( 1 + 43.5 \left( \frac{\mu h_g B \Delta T_{sub}}{\lambda g \Delta T \cdot Pr_{liq}} \right)^{1/2} \left( 1 + \frac{\Delta T_{sub}}{\Delta T} \right) \left( \frac{\rho \Delta \rho}{\Delta \rho} \right)^{1/8} \right) \quad (1)
\]

Figure 2. HTC at cooling stainless steel sphere in isopropanol (a) and in water (b) at \(\Delta T_{sub} = 20\) K.

To determine the wall superheat at vapor film destruction, two semi-empirical models are proposed for the cases of cooling in both saturated and subcooled liquid (equations (2) and (3), respectively). These two equations are obtained from a generic one due to different terms of the vapor film thickness.

\[
T_{w}^{\prime} - T_{lim} = C_0 \cdot \frac{h_g^{3/4} \cdot R_z}{(pc\lambda)_w^{1/2}} \left( \rho' \lambda' \left( T_{lim} - T_j \right) \right)^{1/2} \left( \sqrt{g \Delta \rho} \right)^{1/4} \quad (2)
\]

\[
T_{w}^{\prime} - T_{lim} = C_1 \cdot \frac{\rho' h_g^{3/2} \cdot R_z u_0}{\lambda' \left( T_{lim} - T_j \right) \cdot (pc\lambda)_w^{1/2}} \left( \rho' \lambda' \left( T_{lim} - T_j \right) \right)^{1/2} \quad (3)
\]

Where \(T_{w}^{\prime}\) is the transition temperature (average surface temperature at the vapor film destabilization), \(T_{lim}\) is the limiting attainable temperature of the cooling liquid, \(C_0\) and \(C_1\) are numerical constants fitted to the experimental dataset, \(R_z\) is a surface roughness characteristic size, \(u_0 = \sqrt{g \beta \Delta T_{sub} z / Pr}\) is the characteristic velocity of an upward natural-convective flow, and \(z\) is the linear size of a cooled body.

The resulting temperature head is calculated relative to the limiting attainable temperature of the liquid, which can be evaluated by Snytin's equation [13]. For isopropanol and water at atmospheric
pressure, the corresponding $T_{\text{lim}}$ are 190.6°C and 308°C, respectively. Comparison of the predicted wall superheats with the experimental ones are depicted in figures 3 and 4, respectively, for equations (2) and (3) with empirical constants $C_0 = 0.001$ and $C_1 = 3.65$. The empirical multipliers $C_0$ and $C_1$ are fit to the dataset of 418 experimental transition temperatures (including 57 points for boiling of saturated liquid). The transition temperatures predicted by equation (3) for the experiments depicted in figure 1 are 225°C and 555°C, respectively. It should be mentioned that equation (3) can be applied at $\Delta T_{\text{sub}} < 10$ K, but the probability of incorrect result will increase.

Figure 3. Calculation results by equation (2): ethanol/water-s. steel (1), ethanol-s. steel (2), isopropanol-s. steel (3), water-nickel (4), water-s. steel (5).

Figure 4. Calculation results by equation (3): ethanol/water-duralumin (1), ethanol/water-s. steel (2), ethanol-copper (3), ethanol-duralumin (4), ethanol-gold (5), ethanol-nickel (6), ethanol-s. steel (7), ethanol-silver (8), isopropanol-s. steel (9), water-gold (10), water-nickel (11), water-s. steel (12), water-silver (13).

The scatter of the points in figures 3 and 4 inevitably occurs due to significant uncertainty of the experimental transition temperature. According to the depicted figures, equation (2) for saturated liquid describes 95% of the data within the error band of ±30%, and equation (3) for subcooled liquid describes 78% of the data within the same error band.

3. Numerical simulation: natural convection near the vapor film
During film boiling in subcooled liquid, an upward natural-convective flow inevitably occurs near the vapor film. In this case, the vapor/liquid interface should be considered a free surface due to significant difference in densities of liquid and its vapor.
To check the correctness of the natural convection contribution in equation (3), a numerical simulation of a simplified problem was performed by means of CFD-code ANES20XE, developed in NRU "MPEI".

3.1. Description of the problem
The problem to determine the fields of velocity and temperature at natural convection near the vapor film in the rectangular vessel filled with water is considered. Water bulk temperature is 70°C, pressure above the liquid is atmospheric, and properties of the media are variable. The flow is laminar similar to papers [11, 12], where the assumption on laminar flow was based on visual smoothness of the vapor film during experiments.

The problem is solved in a two-dimensional approximation. The rectangular vessel is modeled by the computational domain of 100×300×40 mm. The right boundary of the computational domain is a wall of the vessel with the no-slip boundary condition for velocity and constant temperature $T_\infty = 70°C$. The top boundary of the computational domain is a free surface which means that the vessel is open, and the bottom boundary of the domain is the bottom of the vessel with the no-slip boundary condition. The left boundary is an axis of symmetry. On this boundary, the vertical plane with a length of 40 mm is located. The lower edge of the plane is at a height of 220 mm from the bottom of the vessel. The slip boundary condition is set on the surface of the plane with constant temperature $T_w = 100°C$.

An unstructured mesh with 80×150×1 control volume with a maximum level of fragmentation 4 is used (figure 5).

Comprehensive descriptions of the mathematical models and numerical methods are given in the user’s manual of the ANES20XE code. Mathematically the problem is described by the system of the conservation laws written in Boussinesq approximation for two-dimensional domain. The following settings were accepted in the code for the simulation: the SIMPLE algorithm to solve Navier-Stokes equations, the KIVA_ILU linear solver, FVES_SUPERBEE type of the numerical scheme, and the convergence criterion is the mean residual with accuracy $10^{-5}$. The calculation converged for every parameter.

3.2. Numerical simulation results
The computational results are depicted in figure 6. According to the obtained velocity field, the longitudinal velocity near the center of the plane is 49 mm/s. This result is in good agreement with calculation by the analytical formula for $u_0$. At similar conditions and at the calculation of the required thermophysical properties by the saturation temperature, the theoretical velocity equals 53 mm/s.
Conclusions
Two equations for wall superheat evaluation at vapor film destabilization have been proposed.

The equation for a saturated liquid boiling with empirical constant \( C_0 = 0.001 \) describes 95% of the 57 experimental points with an error of \( \pm30\% \). The dataset includes data about boiling of saturated and weakly subcooled water, ethanol, ethanol/water mixtures and isopropanol at cooling superheated spherical and cylindrical test samples of stainless steel and nickel at pressures of 0.1–0.3 MPa.

The equation for a subcooled liquid boiling with empirical constant \( C_1 = 3.65 \) describes 78% of the 361 experimental points with an error of \( \pm30\% \). The dataset includes the results on boiling of subcooled water, ethanol, isopropanol and ethanol/water mixtures at cooling superheated spherical and cylindrical test samples of nickel, stainless steel, copper, silver, gold and duralumin at pressures of 0.1–1.0 MPa.

The numerical simulation of the problem of laminar natural convection near the vapor film in subcooled water has been performed. According to the obtained velocity fields, the contribution of the natural convection to the equation for subcooled liquid boiling has been counted correctly.

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