Experimental study of hydrodynamics and heat transfer in two-phase natural circulation loop with reference passive cooling systems of nuclear power plants

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Abstract. In spite of long year practical use of natural circulation systems and not less long period of study working characteristics of these systems the problem of reliable predicting calculations of natural circulation loops (NCL) with boiling coolant at low reduced pressures is not likely can be considered as being solved. At the same time the interest to solution of this problem arises today, because low pressure natural circulation loops are considered as main type of post accident passive cooling systems for nuclear power plants. The main difficulties for calculations are concerned with determining friction pressure losses, which are strongly depended on two-phase flow pattern. At low reduced pressures large difference between liquid and vapour (gas) specific volumes predetermines the strong change of flow pattern even at small change of mass flow quality. An attempts to calculate local two-phase flow parameters (void fraction, circulation velocity) in the wide range of mass flow qualities without considering the change of flow patterns often lead to large deviations of calculated values from the results of experimental measurements. Another specific feature of two-phase low pressure NCL is hydrodynamic flow instability with circulation velocity pulsations of high amplitude and the occurrence of reverse flows. This also presents problems in developing calculation method. In present paper a method for calculating a low-pressure NCL has been developed, in which local two-phase flow parameters (void fraction, the phases velocities, pressure) are calculated using a modified homogeneous model with taking into account the distribution factor and the phases slip and a model of an annular-dispersed flow with considering droplets entrainment and deposition. A modified homogeneous model was applied for description quasihomogeneous flow regimes. At high values of void fraction annular-dispersed flow model was used. Recommendations for change from one model to another in practical calculations have been formulated and verified. The proposed calculation method has been verified by the comparison the calculated thermo-hydraulic characteristics of laboratory natural circulation loop with the experimental data. The experiments have been carried out for boiling in the loop of three different liquids – water, ethanol and perfluorohexane (FC-72) at atmospheric pressure. The comparison of calculated and experimental results showed their good agreement.

1. Introduction. Natural circulation loops (NCL) are in different technologies for a long time. In recent years after the accident at the Fukucima nuclear power plant (2011, Japan), NCL became an object of raised interest as the main type of passive post-accident cooling system of a nuclear reactor core, capable to operate in case of failure or absence of power supply sources and in other extreme
situations. Since the radioactive decay heat removed by the circulating boiling coolant is discharged into the atmosphere, the loop operates at atmospheric (or close to it) pressure. In this connection the number of publications directly focused on the study of NCL operation low pressures has been increased appreciably [1,2]. At the same time, despite the long-term (for several decades) studies of natural circulation loops and their practical use in various technological processes, one still meets both a lack of experimental information about the flow and heat transfer at low pressure natural circulation flow boiling and difficulties in carrying out engineering calculations.

Existing methods for design engineering calculations of NCLs relate to quite high pressures of more than 1 MPa and, in the absence of a rigorous theory, the above calculation rely on semi-empirical techniques based on a homogeneous model. At low pressures, they give large errors in determining flow parameters, which are characterized by large specific vapour volumes and a strong relationship between hydrodynamic and thermal parameters, causing flow instabilities and heat transfer.

One of the main features of the NCL is that the flow rate in it is not holded, but establishes itself in accordance with the operating parameters (thermal load, subcooling of working liquid, pressure), physical properties of the liquid, loop geometry. The disposable pressure difference and hence circulation velocity and temperature regime of the loop establish themselves as the result of interconnection of inner characteristics of the loop. The most important of these characteristics are void fraction and flow pattern. For the conditions of natural circulation knowledge of $\phi$ is more critical In comparison with forced flow conditions, since it determines the driving hydrodynamic head and, therefore, the circulation velocity in the loop. The obvious interdependence of void fraction and circulation velocity multiply complicates the NCL calculation procedure in comparison with two-phase forced flows. At low reduced pressures large specific volumes of the vapor phase causes a rapid growth of volumetric flow quality ($\beta$) and void fraction ($\phi$). this in turn causes a two-phase flow pattern change along the channel height under the conditions of slight change in mass flow quality ($x$). The variety of realizable flow regimes additionally complicates the accuracy of NCL calculations, due to the lack of well-established quantitative criteria to determine the boundaries of the transition between flow regimes.

Among the numerous computational models for calculating friction in two-phase flows, only the homogeneous one for flows of the quasi-homogeneous structure [3] and annular flow model by Wallis [4] are based on clear physical concepts. The homogeneous model is not only the oldest and simplest one of the proposed models of two-phase flows, but objectively far from the worst, since it is sufficiently physically justified. Examples of good agreement of friction factors calculated according to homogeneous model and experimental data for steam-water flows at pressure of 4.9 MPa are given in [5]. At low reduced pressures, the applicability of a homogeneous model is limited to very low mass flow qualities. Here in most cases they deal with flows of annular structure. Since the determining of boundaries of various two-phase flow patterns in calculations is more complex problem than pressure gradient calculations, it seems reasonable to consider two enlarged flow patterns — a quasi-homogeneous (bubble, slug, churn) one and separate one. In the case of vertical channels this is annular-dispersed flow. Such an approach is justified in [3] and is largely consistent with the analysis of the review report [6]. The possibilities of applying the homogeneous and annular-dispersed flow models to the analysis of experimental data on the characteristics of two-phase flows at low pressures are shown in [7]. Taking as a basis an analysis of the available experimental data on characteristics of adiabatic two-phase flows, the authors have shown that the use of homogeneous model for quasi-homogeneous flow regimes and annular flow model at high values of void fraction gives satisfactory results. The necessity of taking into account the flow structure (flow regime) when calculating friction and void fraction is substantiated. A method for determining the transition boundary from one model to another in practical calculations has been proposed and verified.

Taking into account the results obtained in [7], the present paper proposes a method for calculating low-pressure NCLs, in which local two-phase flow parameters (void fraction, phase velocities, pressure) are calculated on the basis of the modified homogeneous model and the models of a dispersed-annular flow. The modified homogeneous model takes into account the distribution factor
and phase slip according to Labuntsov’s model [3,8]. In the model of a dispersed-annular flow entrainment and deposition of droplets are taken into account. [9]. The proposed calculating method was verified by comparison with experimental data obtained in the laboratory natural circulation loop [10]. A set of experimental data for water and perfluorohexane boiling obtained by the authors earlier [10] and new data for ethanol boiling have been used for the comparison. Detailed description of the experimental data, their analysis and qualitative classification of flow and heat transfer regimes from the standpoint of the stability of circulation, characteristics of wall temperature pulsations are given in [10]. The use of three liquids with significantly different properties, expands the range of the given reduced pressures and, therefore, realizable flow regimes. At the same time, quite satisfactory results of comparing the calculation method with the experimental data obtained on different liquids makes it possible to draw a conclusion about the universality of the proposed approach.

2. Experimental apparatus. The detailed description of experimental natural circulation loop and measuring procedure are given in [10]. A schematic diagram of the loop is presented in Figure 1. The flow-up channel and down-comer were connected to the separator-condenser mounted at the top of the loop. The down-comer was supplied by the cooling heat exchanger and two electrical heaters for predetermining and maintaining the liquid temperature at the inlet to the heated section. Flow-up (test) section was a thin-wall stainless steel circular tube with inner diameter of \( d = 9.1 \text{ mm} \) which consisted of two parts – the upper adiabatic section 48 diameters long and lower heated section 98 diameters long. The test section was heated by stabilized alternating current, which traversed immediately the tube wall.

Measured quantities in the experiments were heat power supplied to the test section, working liquid flow rate, liquid temperature at the inlet to the flow-up section, temperature distribution along the heated wall, pressure in separator-condenser chamber. The flow rate in the loop was measured with ultrasonic flow meter FLUXUS F601 with the error of 0.01 \( \text{m/s} \). The pressure transducer AIR of 0.2% accuracy class was used for pressure measuring in the separator-condenser chamber. Temperature measurements are in error by 0.2 \( \text{oC} \), heat flux density measuring error is 2%.

3. The proposed method for hydraulic calculation of natural circulation loop at low pressures.

3.1. Calculation of friction and void fraction. The hydraulic calculation of the natural circulation loop is based on the equality of the driving head to the sum of the hydraulic losses in the entire loop:

\[
\Sigma \Delta P_{\text{loss}} = \Delta P_{\text{dh}}.
\]

(1)

Total pressure losses include friction losses, acceleration, and local losses throughout the circuit. The difference in the densities of the medium in flow-up and flow-down legs produces a driving head, which for vertical channels is determined from the relationships,

\[
\Delta P_{\text{dh}} = g (\rho' - \bar{\rho}_{\text{mix}}) h,
\]

(2)

\[
\bar{\rho}_{\text{mix}} = (1 - \bar{\phi}_\rho \rho' + \bar{\phi} \rho^v),
\]

(3)

where \( h \) is the height of the pipe section, \( \bar{\rho}_{\text{mix}} \) – the average actual density of the medium (liquid-vapour mixture) in this section, \( \bar{\phi}_\rho \) – the length-averaged void fraction, \( \rho' \) is liquid density, \( \rho^v \) is vapor density.
The aim of the calculation is to determine the circulation velocity \( w_0 \) (mass average velocity of a single-phase liquid at the inlet to the heated zone) at which equation (1) is satisfied. The input parameters of the calculation are wall heat flux density \( q \) and the temperature of liquid at the inlet to the flow-up heated section \( T_{\text{in}} \).

The single-phase convective heat transfer in the heated zone is calculated from the relations for co-current effect forced and natural convection. Given the known heat balance, the bulk temperature is determined, the heated section wall temperature \( T_w \) is determined if the heat transfer coefficient (HTC) is known, and the variation of local pressure \( p_{\text{loc}} \) is calculated. When certain wall superheating is reached, boiling starts, and a shift is made for calculating a two-phase flow. In the two-phase region, vapor quality \( x \), vapor volumetric flow ratio \( \beta \), and void fraction \( \phi \) over the heated zone height are added to the main calculated characteristics mentioned above. Based on the void fraction distribution along the channel axis \( \phi(z) \) and flow pattern changing criteria, the flow cross-section separating the quasi-homogeneous and annular flow regimes is determined. The heat transfer and flow hydrodynamic parameters in each regime are calculated using different relations. The vapor quality in each section \( x(z) \) is determined from the thermal balance equation

\[
x(z) = \frac{\pi d q}{G_{\text{mix}} h_{\text{LG}}} (z - z_0),
\]

where \( d \) is the channel inner diameter, \( G_{\text{mix}} \) is the mixture mass flow rate, \( h_{\text{LG}} \) is the heat of vaporization, and \( z_0 \) is the longitudinal coordinate boiling incipience flow cross-section.

For steady-state regimes, which will later be used for comparison with calculations, the pressure drop in the channel axis direction (\( z \)) for a one-dimensional two-phase flow is described by the equation

\[
-\frac{dp}{dz} = \frac{G_{\text{mix}}}{S} \frac{d}{dz} \left[ w'x + w'(1 - x) \right] + \frac{4\tau_{\text{mix}}}{D} + \rho_0 g z,
\]

where \( S \) is cross-sectional area of the channel, \( w' \) and \( w' \) – actual velocities of vapour and liquid, \( x \) – vapour quality, \( \tau_{\text{mix}} \) – wall shear stress, \( G_{\text{mix}} \) – mass flow rate of the mixture, \( D \) – inner channel diameter, \( \rho_0 \) – actual mixture density, \( g_z \) – gravity acceleration vector on \( z \) direction. The right-hand terms of equation (2) express the components of pressure gradient due to flow acceleration, friction on the wall, and mass forces respectively.

As it is seen the main difficulty is the calculations of shear stresses and void fraction because these parameters depend on two-phase flow pattern. As it was pointed out above the main distinction of the proposed approach is in taking into account the real two-phase flow pattern for calculation void fraction and friction. The detailed analysis of this approach and the results of its testing on the basis of experimental researches is described in [7]. General recommendations for calculating pressure drop and void fraction in two-phase flow and design-basis criteria of transition one flow pattern to another are given below.

For vapour-liquid flows at low reduced pressures due to large specific vapour volumes, the volumetric rate flow quality (\( \beta \)) and void fraction (\( \phi \)) are growing rapidly with a slight change in the mass-flow quality (\( x \)). An example of calculation of these parameters for vapour-liquid flows of water, ethanol and perfluorohexane at atmospheric pressure for particular regime parameters in the loop (\( q = 64 \) kW/m\(^2\), \( T_{\text{in}} = 97 \) °C, \( q = 54 \) kW/m\(^2\), \( T_{\text{in}} = 78 \) °C, \( q = 30 \) kW/m\(^2\), \( T_{\text{in}} = 54 \) °C correspondingly) is given in figure 2.

It follows from figure 2, a,b that in water and ethanol vapour-liquid flows very low mass flow qualities (\( x \geq 0.01 \) and \( x \geq 0.022 \) correspondingly) identify a condition \( \beta \geq 0.94 \), that is as a rule makes one to suggest annular-dispersed flow regime for two-phase mixture. In two-phase perfluorohexane flow one observes not so rapid \( \beta \) increase than in water and ethanol (figure 2, b).
a) - water at \( q = 64.3 \text{ kW/m}^2, T_{in} = 97.3 \degree\text{C} \); b) - ethanol at \( q = 54.3 \text{ kW/m}^2, T_{in} = 78 \degree\text{C} \); c) - FC-72 at \( q = 30 \text{ kW/m}^2, T_{in} = 54 \degree\text{C} \).

2 - calculation according to homogeneous model with account of phase slip, 3 - calculation according to annular flow model, 4 - calculation according to annular flow model with account of droplets entrainment and deposition.

**Figure 2.** Change of volumetric flow quality \( \beta (1) \) and void fraction \( \phi (2-4) \) with mass flow quality (along the height of the flow-up section for boiling of different liquids.

At these regime parameters \( (q = 30 \text{ kW/m}^2, T_{in} = 54 \degree\text{C}) \) bubble flow regime will probably mainly prevail. As one can see in figure 2, b volumetric rate flow quality only at the end of the heated section becomes equal to \( \beta \approx 0.894 \) at mass flow quality at \( x = 0.07 \). This is caused by practically an order of magnitude larger reduced pressure for perfluorohexane than for water (the same difference takes place for vapour to liquid ratio for these liquids).

The value \( \phi = 0.7 \) is usually taken as the annular structure existence lower boundary. Therefore, the calculation according to the homogeneous flow model is carried out at \( \phi_{ann} \geq 0.7 \), and then the calculation should be carried out according to both the homogeneous model and the dispersed–annular flow model after reaching this value. Given the same flow parameters, the void fraction value calculated using the dispersed-annular flow model is lower than that determined according to the homogeneous model, as is seen from figure 2. A comparison of the calculation results with the measured friction values for forced two-phase flow has shown that the change from homogeneous to annular flow model should be done when calculated void fraction reaches the value \( \phi_{ann} = 0.7 \) according to annular model.

In figure 2, a for water the value \( \phi_{ann} = 0.7 \) corresponds to \( \phi_{hom} \approx 0.816 \) at \( x = 0.012 \) and \( \beta \approx 0.948 \).

Similarly for ethanol and perfluorohexane (figures 2, b,c) the value \( \phi_{ann} = 0.7 \) corresponds to \( \phi_{hom} \approx 0.805 \) \((x = 0.0255 \text{ and } \beta \approx 0.932)\) and \( \phi_{hom} \approx 0.769 \) \((x = 0.070 \text{ and } \beta \approx 0.894)\) respectively.

On the basis of the analysis made earlier by the authors, in proposed method the friction pressure losses are calculated by the classic homogeneous model \( (\phi = \beta) \), but in driving head calculations the phase clip is taken into consideration according to D.A. Labuntsov’s model [8].

In the framework of a homogeneous model for a hydrodynamically stabilized two-phase flow, the relationship for pressure gradient due to frictional losses is written in the same way as for single-phase flow:

\[
\left(\frac{dp}{dz}\right)_{fr} = \frac{4\tau_{mix}}{d} ; \quad \tau_{mix} = \frac{8}{\rho} \rho_{mix} w_{mix}^2 .
\]  

In the equation (3) the mixture density determined from volumetric rate flow quality (homogeneous mixture density) is \( \rho_{\beta} = (1-\beta)\rho + \beta \rho^* \), and the mixture velocity \( w_{mix} \) is equal to the sum of the reduced liquid and gas velocities, is calculated by the following relationship.
\[
\rho_{\text{mix}} = \rho_{0}\left[1 + x\frac{\rho' - \rho''}{\rho'} \right] = \frac{\rho_{0}}{1 - \beta(\rho' - \rho'')/\rho'}.
\]  

(7)

The void fraction calculation \(\varphi\) is carried out according to the model [8], based on the kinematic phase-slip scheme

\[
\frac{\beta}{\varphi} = C + \frac{\Delta w}{\rho_{\text{mix}}}. 
\]  

(8)

The first term on the right-hand side of equation (8) (the distribution factor) takes into account the inhomogeneity of the velocity and void fraction over the channel cross section. According to [3], for forced flow in channels under moderate and high reduced pressures the distribution factor is \(C = 1.1\).

The second term on the right-hand side of equation (8) takes into account the local phase slip, \(\Delta w\) is the float up \((u_\infty)\) of gas bubbles in a stationary liquid multiplied by the dimensionless parameter reflecting the effect of mutual entrainment of bubbles:

\[
\Delta w = \varepsilon u_\infty; 
\]  

(9)

\[
\varepsilon = 1.4\left(\rho'/\rho^*\right)^{1/5}\left(1 - \rho''/\rho'\right)^{5};
\]

\[
u_\infty = 0.35\sqrt{g d \Delta \rho/\rho^*}.
\]

To calculate the pressure losses in a dispersed-annular flow the Wallis approach based on the analysis of separate flows of liquid and gas (vapour) [4] was used. This analysis was supplemented with an additional model [10] which accounts the flow rate of liquid in the form of droplets dispersed in vapour (gas) core. The void fraction \(\varphi\) in an unique fashion is determined by the thickness of liquid film \(\delta\).

The shear stress on the interface \(\tau_i\) is determined from the average actual gas velocity \(w''\), measured from the liquid velocity on the film surface \(w'\) [9]. The liquid velocity on the film surface \(w'\) is calculated in accordance with the universal logarithmic Prandtl law [14]:

\[
w'_{\delta} = 2.5\ln\left(\frac{\delta/\rho'}{v'}\right)\sqrt{\frac{\tau_{\delta}/\rho'}{\rho'}} + 5.5\sqrt{\tau_{\delta}/\rho'}
\]

\[
\tau_i = \tau_{\delta} + \Delta \rho g d \delta (1 - \delta/d)
\]

(10)

The friction factor on the interface \(\xi\) is determined according to the method described in [9], within the framework of which it is assumed that a liquid film with a wave surface acts as a sand roughness with a grain size equal to the quadruple film thickness.

\[
\xi_i = \xi_0 \left(1 + \frac{240\delta}{d}\right)
\]

(10,a)

The numerical coefficient is adjusted according to [3], and \(\xi_0\) is friction factor calculated for gas flow in a hydraulically smooth pipe with a diameter \(d_i = d - 2\delta\). The parameters of the two-phase flow obtained within the framework of the annular model are corrected by taking into account the liquid droplets entrainment to the gas core by the method [9].

The model [9] refers to low reduced pressures. At the same time it is considered the channel to be long enough so the equilibrium between the droplets entrainment flow density from the film surface \(E\) and back flow of droplets deposition \(D\) has time to set in. The final balance equation \(E = D\) from the work [9] can be written as follows
In this equation \( \Gamma_F = \frac{m_F}{(\pi d)} \) – is specific liquid flow rate; \( m_G, m_F, m_E \) – flow rates of gas, liquid in film, liquid in flow core in droplets (kg/s) respectively; \( m_F + m_E = m_L \) – total liquid flow rate; \( \xi^* \) – friction factor of gas flow, being determined by its reduced velocity.

Strictly speaking the equilibrium condition of mass flow density of droplets entrainment and back flow of droplets deposition was hardly achieved in all experiments which results have been used for comparison with calculations. Nevertheless the account of droplets deposition in calculations gives more realistic results both for film thickness and pressure gradient. This account leads to a decrease in film thickness and corresponding correction of pressure loss component due to friction.

The two-phase flow parameters obtained using the annular model are corrected taking into account droplet entrainment of liquid into the gaseous core according to the procedure outlined in [9], which primarily results in that the film thickness becomes thinner and a relevant correction is made to the friction loss component. The model of a dispersed-annual two-phase flow in a vertical channel is described in detail in [7].

3.2. Determining the boiling incipience cross-section and calculation of heat transfer at the heated section of the loop. At low reduced pressures, the single-phase convection zone can be of considerable length, and therefore the accuracy of the NCL calculation depends significantly on the degree of reliability of determining the cross-section coordinate of boiling incipience since this determines the correctness of calculating the void fraction in heated section and the most important parameter – the circulation velocity of the coolant as the result.

To determine the temperature difference of boiling incipience \( \Delta T_{bi} = T_w - T_s \) a simple model based on experimental observations has been used. The minimal (equilibrium) size of a spherical vapor nucleus is determined from the Laplace equation, which gives the following expression after approximate replacement of pressure difference by temperature difference according to the Clausius–Clapeyron equation:

\[
0.014\pi d \left( \frac{\mu^3 \Gamma_F^3 w^2}{\sigma^2} \right)^{1/4} = \frac{\xi^*/8}{\left(1 - 12.7 \sqrt{\xi^*/8} \right)} \left( \frac{m_G m_E}{m_G + m_E} \right)
\]

(11)

where \( \sigma \) – is surface tension.

Expression (12) determines the size of a viable vapor nucleus only if there is a uniform temperature field in liquid. If a bubble is generated on a solid surface, there is a temperature gradient inside the thermal boundary layer, which is directed normally to the surface. According to the model presented in [11], the necessary superheating of liquid should exist at a distance from the wall larger than \( R_f \). In view of smallness of this size, we can assume that the temperature profile at the wall is linear and adopt the following expression correlating the temperature difference in the layer of thickness \( \delta_T \) with the heat flux:

\[
q = \frac{\lambda \Delta T}{\delta_T}
\]

(13)

where \( \lambda \) – is thermal conductivity of liquid. Supposing the thickness of this overheated liquid layer to be equal to vapour nucleus size \( \sigma_T = R_f \) one can find a relation between temperature difference of boiling incipience, heat flux density and liquid properties.

Simultaneous solution of the equations for vapour nucleus radius and temperature gradient in thermal liquid layer in the direction normal to the heated surface gives the expression for \( \Delta T_{bi} \).
\[ \Delta T_{\text{bi}} = \frac{C_1}{1 - \cos^2 \Theta} \sqrt{\frac{\sigma T q}{h_{\text{LG}} \rho' \nu}} \]

where \( \Theta \) is the contact angle.

The constant \( C_1 = 2.5 \), which is a function of contact angle \( \Theta \), was selected based on experiments with three liquids. That the surface wettability has an effect on the liquid boiling incipience is a well-established fact. With \( \Theta = 78^\circ, 30^\circ \), and \( 20^\circ \) for water, ethanol, and perfluorohexane, respectively. The numerical values are consistent, at least qualitatively, with the characteristics of metal surface wettability with these liquids.

In single-phase convection zones in mixed circulation regimes the wall temperature is determined by the intensity of heat transfer in single-phase convection: \( T_w(z) = \bar{T}(z) + \frac{q_w}{\alpha_w} \), where: \( \alpha_w \) is the heat transfer coefficient, calculated from one of the formulas for single-phase heat transfer, \( T(z) \) is bulk liquid temperature.

To date, for engineering practice, no justified recommendations have yet been proposed for the calculations of heat transfer and friction for single-phase liquid flows in systems with natural circulation, so in this paper calculations of friction and heat transfer in zones to boiling have been performed according to B.S. Petukhov’s recommendations for flows with the same direction of forced and natural convection [12, 13].

With boiling onset the calculation of the heat transfer coefficients in two-phase flow region is carried out according to the interpolation relation, taking into account the joint effect of the heat transfer mechanisms due to the convective component and vaporization

\[ \alpha = \left[ \left( 1 + \frac{\Delta T}{T_w - T_b} \right)^{3/2} + \alpha_q^3 \right]^{1/3} \]

where \( \alpha_w \) is the heat transfer coefficient for single-phase convection; \( \alpha_q \) is the heat transfer coefficient at bubble boiling, which is calculated according to [14].

\[ q = 3.43 \cdot 10^{-3} \frac{\lambda^2 \Delta T}{v \sigma T_s} \left( 1 + \frac{h_{\text{LG}} \Delta T}{2 R T_s} \right) \left( 1 + \sqrt{1 + 800B + 400B} \right), \quad B = \frac{h_{\text{LG}} (v \rho' \nu)^{1/2}}{\sigma (\lambda T_s)^{1/2}} \]

In a two-phase flow the wall shear stress greatly increases with void fraction increasing. In certain conditions the intensity of convective heat transfer behaves in the same way. This analogy of momentum and energy transfer in a homogeneous two-phase flow was used in the analysis of heat transfer during condensation in pipes [13]. In the work of L.D. Boyko and G.N. Kruzhilin it is concluded that for non-metallic liquids with \( Pr > 1 \) the main thermal resistance is concentrated in the heat-conducting sublayer, since in the case of turbulent flow beyond the viscous sublayer the Reynolds analogy for momentum and energy transfer is valid.

\[ \alpha \approx \frac{\lambda}{\Delta \lambda} \approx \frac{\lambda P r^{1/3} \nu_s}{v} ; \nu_s = \sqrt{\tau_w / \rho} \]

From this it follows the conclusion that the heat transfer coefficient is proportional to the square root of wall shear stress: \( \alpha \sim \sqrt{\tau_w} \). It follows from this considerations that the the convective component of heat transfer coefficient for flow boiling in the channels can be calculated as [15].

\[ \frac{\alpha_{\text{mix}}}{\alpha_0} = \frac{\tau_{\text{mix}}}{\tau_0} \]

Wall shear stress in two-phase flow regimes \( \tau_{\text{mix}} \) is calculated according to correspondent correlations for homogeneous or annular-dispersed flow model; \( \alpha_0 \) and \( \tau_0 \) are heat transfer coefficient and wall shear stress in single-phase flow.
The combination of relations (1) – (16), together with the equations for single-phase heat transfer forms a closed system of algebraic and transcendental equations for a known circulation rate \( w_0 \). This system was solved using standard PC programs. The circulation rate was calculated iteratively to the coincidence of the values of the driving head and total pressure losses in the loop with a given accuracy, the heated zone being partitioned into control volumes and calculations being performed step by step for each control volume.

4. Comparison of experimental data with the calculation. Practical calculations have been performed using an iterative procedure implemented as a computer program. In the course of a single iteration, the flow and heat transfer modes over the length of the heated zone are diagnosed for a given circulation rate, the friction factors, the longitudinal variation of the wall temperature, and the longitudinal variation of void fraction in two-phase flow zone have been calculated. The iterative procedure is terminated when the design circulation rate assumes a value that satisfies the driving head equality to the total pressure losses in the loop.

Calculation results have been compared with the experimental data, which were obtained on three different liquids in the ranges of heat flux densities: (11 – 74) kW/m² for water, (20 – 80) kW/m² for ethanol and (5 – 30) kW/m² for perfluorohexane at atmospheric pressure. The ranges of liquid subcooling to saturation temperature were respectively: \((0 – 70)\) K, \((0 – 5)\) K and \((3 – 15)\) K.

Significant difference in physical properties of liquids made it possible to expand the range of research on reduced pressures by almost an order of magnitude \(\left(\frac{p}{p_{cr}} = (0.475 - 5.5) \times 10^{-2}\right)\) without changing the total pressure in the loop. This in turn gave one an opportunity to extend the set of possible two-phase mixture flow patterns in heated section of the loop and to obtain an additional information to analyze the influence of a two-phase flow pattern on mechanism of circulation instability and the conditions for its occurrence.

The occurrence of hydrodynamic instability is characteristic of any two-phase flows, but in natural circulation loops this problem becomes noticeably aggravated, since the circulation rate is not set by the injection device, but is a function of the process. This is especially true for the low-pressure NCL due to the large values of vapor phase specific volume. In particular, the role of a well-known type of instability due to the onset of boiling increases. The appearance of vapor phase leads to a sharp decrease in the density of the medium in the heating zone, i.e. to the growth of the driving head and circulation rate. The growth of the flow velocity suppresses the boiling process, the density of the medium increases, the driving head and the circulation rate decrease. Naturally, the intensity of heat transfer decreases, the wall temperature rises, this, some time later, produces conditions for boiling again. The process is repeated. With its computational simulation, even determining the local saturation temperature becomes problematic.

The liquid subcooling to saturation temperature at the inlet to the heated section, at low wall heat flux densities leads, as a rule, to instability of circulation in the loop. In boiling flows the fluctuations of wall temperature and flow rate are unavoidable; so a stable circulation regime should be understood only as one in which these fluctuations are below a certain threshold level. As in the case of turbulent flow, the characteristic values of the velocity \(w\) and wall temperature \(T_w\) in NCL should be their values averaged over a certain period of observations, although the nature of the pulsations here is completely different.

The proposed calculation method does not consider the analysis of non-stationary processes. Nevertheless, a calculated circulation velocities and wall temperatures have been compared with the experimental data obtained not only at stable circulation regimes, but also at regimes characterized by instability with significant fluctuations of the flow rate and wall temperature. In the last case averaged over the ensemble values of \(T_w\) and \(w_0\) have been used.

As an example, figure 3 shows calculated and averaged experimental values of circulation velocity \(w_0\) for water at \(q = 45\) kW/m² as depended on the liquid temperature at the inlet to the heated section.
Figure 3. Experimental and calculated circulation velocities as depended on inlet temperature of liquid for water at \( q = 45 \text{ kW/m}^2 \)

Vertical straight-line segments in figure 3 show the standard deviations of circulation velocity fluctuation amplitudes from corresponding mean value.

As liquid subcooling increases, the amplitude of oscillations may exceed twice the averaged value. At the same time in some regimes reverse flows have been observed and in flow meter time records flow velocity passes through zero into interval of negative values. As subcooling decreases the amplitude and period of oscillations decrease, and the discrepancy between the averaged experimental velocity and calculated one decreases down to 18\% (\( \Delta T_{\text{sub}} = 5 \text{K} \)). In this case, the use of the latter in the calculations of wall temperatures \( T_w \) makes it possible to obtain good agreement with the measurement results – see figure 4.

Figure 4. Wall temperature distribution along the height \( z \) of test section for water boiling at \( q = 45 \text{ kW/m}^2 \)

A typical time records of wall temperature and circulation velocity for the regime with flow instability, fairly long zone of single-phase convection and irregular vaporization at the end of the heated section are shown in figure 5, a,b. The oscillograms of the circulation velocity and wall temperature at six distances from the inlet to the heated section are shown for water circulation at wall heat flux density \( q = 10 \text{ kW/m}^2 \) and inlet liquid subcooling \( \Delta T_{\text{sub}} = 9 \text{ K} \).
Figure 5. Measured and calculated parameters for water boiling at \( q = 11 \text{ kW/m}^2 \) and \( T_{\text{in}} = 90^{\circ}\text{C} \)

Figure 5, c shows a comparison of the ensemble-averaged values of measured wall temperature with calculated curve.

Analysis of wall temperature oscillograms in those sections where the wall temperature periodically exceeds the value of boiling incipience point, allows one to draw the following sequence of events.

When boiling starts the driving head and the flow rate in the loop increase. As a result, convective heat transfer intensifies and wall temperature decreases; this suppresses boiling, and driving head and loop flow rate reduce. After that, the wall temperature rises again, boiling occurs – the process repeats. The synchronism of such oscillations in channel cross-sections at different distances from the inlet causes large fluctuations of void fraction in heated section. The frequency of temperature fluctuations correlates with circulation velocity fluctuations and is of the order of 15-16 s.

As it has been revealed in water experiments that in this case the circulation instability occurs in regimes with high subcooling of working liquid at the inlet to the heated zone. This regimes are characterized by highly long zone of single-phase convection. Under these conditions the hydrodynamic flow pulsations in the loop are caused by the longitudinal displacement of boiling incipience flow cross-section. Mixed flow regimes with extended zones of single-phase convection and boiling (low wall heat flux densities, large inlet liquid subcooling) are characterized by instability of the flow and heat transfer.

As wall heat flux density increases and inlet liquid subcooling decreases the amplitude of the temperature pulsations decreases, and their frequency increases. When the length of the single-phase zone becomes small compared to the total height of the heated section, it is possible to speak about the onset of a steady-state NCL operation, although this definition is arbitrary, since it is impossible to completely eliminate fluctuations in wall temperature under conditions where the circulation rate depends on the process. This is clearly seen from figure 6, which shows the regime for water boiling at \( q = 64 \text{ kW/m}^2 \) and \( \Delta T_{\text{sub}} = 3 \text{ K} \).
Figure 6. Time records of wall temperature (a), circulation velocity (b) and wall averaged temperature distribution along the height of the heated section (c) for water boiling at $q = 64.3 \text{ kW/m}^2$ $T_{in} = 97.3^\circ \text{C}$. Legend is the same as in figure 5.

The amplitude of velocity pulsations is at the level of 10% of its average value, and for local wall temperatures did not exceed 2 K, the nature of their oscillations being similar to that observed at bubble boiling. This mode can be considered as stable, bearing in mind the small amplitude of circulation velocity and wall temperature oscillations. Experimental and calculated wall averaged temperature distribution along the heated section are shown in figure 6, c. The calculated $T(z)$ dependence has two non-monotonicities, the first of them reflects the onset of boiling, the second one – the transition from homogeneous flow pattern to annular-dispersed flow (figure 6, c. The largest deviation of averaged measured wall temperature from calculated one is 3 K. In experiments on water, the region of stable circulation of boiling coolant occurred only at wall heat flux densities above 60 kW/m$^2$.

Studies on non-aqueous liquids (perfluorohexane, ethanol) were carried out at low subcoolings (0 - 15) K. Experiments have shown that in all boiling regimes of perfluorohexane (figure 7) and in boiling regimes of ethanol at $q < 30 \text{ kW/m}^2$ (figure 8) the flow and heat transfer were stable. For all modes the temperature fluctuations of the heating wall were insignificant or practically absent, although the velocity fluctuations took place, their amplitude does not exceed 10% of the averaged value of circulation velocity. The calculation method describes well the experimental data on longitudinal wall temperature distribution.

Figure 7. Time records of wall temperature (a), circulation velocity (b) and wall averaged temperature distribution along the height of the heated section (c) for FC-72 boiling at $q = 30 \text{ kW/m}^2$ $T_{in} = 54^\circ \text{C}$. Legend is the same as in figure 5.

The calculated $T_w(z)$ curves for all three liquids studied have in general two non-monotonicities, the first of which reflects the onset of boiling, and the second one – a possible transition from a homogeneous to annular-dispersed flow pattern. The longitudinal coordinates of boiling incipience flow cross section correspond to the sharp peaks on $T_w(z)$ curves in accordance with $\Delta T_{bi}$ values.
calculated by the equation (15). These wall overheating values have been are \((1–2)\) K for water \((q = 20 – 100\ kW/m^2)\), \((3–7)\) K for ethanol \((q = 20 – 80\ kW/m^2)\), \((4–9)\) K for perfluorohexane \((q = 5 – 30\ kW/m^2)\). In the majority of the regimes considered the calculated coordinate of the boiling incipience cross section quite satisfactorily coincides with the results of measurements (see figures 4, b,c, figure 6, c (water); figure 7, c (perfluorohexane)). When ethanol boils at high heat fluxes, the above peaks coincide with the measurement results only qualitatively.

**Figure 8.** Time records of wall temperature (a), circulation velocity (b) and wall averaged temperature distribution along the height of the heated section (c) for ethanol boiling at \(q = 30\ kW/m^2\) and \(T_{in} = 76^\circ\ C\). Legend is the same as in figure 6.

**Figure 9.** Time records of wall temperature fluctuations (a) and wall averaged temperature distribution along the height of the heated section (b) for ethanol boiling at \(q = 54.3\ kW/m^2\) and \(T_{in} = 78^\circ\ C\). Legend is the same as in figure 5.

Small protrusions (by 1-2 K) on the calculated \(T_w (z)\) dependences due to the change in two-phase flow patterns visible in figure 6, c (water) and figure 9, b (ethanol). As liquid film evaporates the heat transfer coefficient increases and the calculated wall temperature decreases along the length of the channel. The agreement with the results of the experiment is preserved both in the calculation by the homogeneous model and in the change over to the calculation by annular-dispersed flow model.

A change in the two-phase flow, both in the considered examples and in the other modes studied, does not cause a significant change in the calculated wall temperature. However, the transition to annular-dispersed flow model in the calculations means a reduction in friction losses by a factor of 2-3 compared to homogeneous model and significantly increases the calculated circulation rate.

When ethanol boils at heat flux densities \(q \geq 30\ kW/m^2\), significant temperature fluctuations have been observed for ethanol boiling at heat flux densities \(q \geq 30\ kW/m^2\) (see figure 9, a). Analysis of the calculation results of local flow parameters leads to the conclusion that in this case the most likely cause of instability in the loop is a periodic change of flow regimes in the heated section. Significantly lower heat of evaporation of ethanol compared to water and less reduced pressure compared to perfluorohexane means higher values of mass flow quality at identical heat fluxes and an earlier transition to annular-dispersed flow (figure 9, b). According to calculations, the transition to annular-
dispersed pattern occurs already at a distance of about 0.5 m (55 calibers) from the inlet to heating section, and at the maximum wall heat flux density \( q = 80 \text{ kW/m}^2 \) the transition to this pattern began after 30 calibers. This transition means an approximately threefold decrease in friction resistance with a relatively small change in the average mixture density, which leads to a significant increase in circulation velocity. The increase in velocity leads to a decrease in steam content and shifts upward the point of transition to the annular flow. This increases friction and circulation velocity decreases. Further the process repeats. It may be a probable cause of instability occurrence during the ethanol boiling in the loop at wall heat flux densities \( q > 35 \text{ kW/m}^2 \).

The generally satisfactory agreement between the measured and calculated thermo-hydraulics characteristics of NCL, indicated in this work, confirms the conclusion [7] of the need to take into account the flow pattern in calculations of two-phase flows.

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