Numerical simulation of velocity and temperature fields in natural circulation loop

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Abstract. Low flow natural circulation regimes are realized in many practical applications and the existence of the reliable engineering and design calculation methods of flows driven exclusively by buoyancy forces is an actual problem. In particular it is important for the analysis of start up regimes of passive safety systems of nuclear power plants. In spite of a long year investigations of natural circulation loops no suitable predicting recommendations for heat transfer and friction for the above regimes have been proposed for engineering practice and correlations for forced flow are commonly used which considerably overpredicts the real flow velocities. The 2D numerical simulation of velocity and temperature fields in circular tubes for laminar flow natural circulation with reference to the laboratory experimental loop has been carried out. The results were compared with the 1D modified model and experimental data obtained on the above loop. The 1D modified model was still based on forced flow correlations, but in these correlations the physical properties variability and the existence of thermal and hydrodynamic entrance regions are taken into account. The comparison of 2D simulation, 1D model calculations and the experimental data showed that even subject to influence of liquid properties variability and entrance regions on heat transfer and friction the use of 1D model with forced flow correlations do not improve the accuracy of calculations. In general, according to 2D numerical simulation the wall shear stresses are mainly affected by the change of wall velocity gradient due to practically continuous velocity profiles deformation along the whole heated zone. The form of velocity profiles and the extent of their deformation in its turn depend upon the wall heat flux density and the hydraulic diameter.

1. Introduction

It is no need to say that natural circulation loops or loop thermosyphons are heat removal systems which have many industrial applications of different purpose. In recent years natural circulation heat removal mechanism is considered for passive safety systems in new design nuclear power plants. Besides, some advanced nuclear plant designs rely on natural circulation to remove core power under normal operation (startup, normal power operation, and shutdown), and some designs rely on natural circulation to provide cooling of the containment. Many applications including passive safety systems of nuclear power plants operate at low parameters, very often low flow conditions of single phase heat transfer medium are realized. So the availability of the reliable engineering and design calculation methods of flows driven exclusively by buoyancy forces is an actual problem. In particular it is important for the analysis of start up regimes of passive safety systems of nuclear power plants.
Natural circulation systems may have low driving force and need to be started from the state of rest. Start-up from rest is one of the key issues in assessing the reliability of these systems. There is always a finite time lag before these systems attain their optimum intended performance level. The predicting models applicable for low flow conditions can be quite different from those applicable for high flow conditions. For example, in many cases under low flow conditions the three-dimensional flow effects occur and traditional 1D approach may not predict the behaviour of natural circulation systems with reasonable satisfaction. Also the performance of these systems is strongly dependent on the operating conditions and system geometry [1, 2].

Being of great practical importance single-phase natural circulation loops have been investigated experimentally, analytically and numerically for several decades since early 1950s and remains to be the subject of research till today. One of the problems having been not solved yet is the correct friction factor calculation for the conditions of complex velocity fields and great variability of liquid properties in the flows induced exclusively by buoyancy forces. The most complex flow fields are in laminar flow regimes.

In spite of long year investigations of natural circulation loops no suitable predicting recommendations for heat transfer and friction for the above regimes have been proposed for engineering practice and correlations for forced flow are commonly used which to large extent overpredict the real flow velocities [1–5]. In particular the predicting friction factor correlations for fully developed isothermal flows is very often used in calculation of loop thermosyphons [2–4].

To our point of view this approach is not fully correct because the form of velocity profiles in real buoyancy-induced circulation flows differs from fully developed isothermal velocity profiles. Besides the profile configuration can change along the heated zone and it will lead to additional increase of hydrodynamic drag.

The present work was inspired by the results of comparison of 1D calculations with the experimental data on two-phase natural circulation in laboratory loop thermosyphon designed for studying natural circulation of boiling liquid at low pressures [6, 7]. As at low pressures boiling the single-phase convection region can be relatively very large low flow experiments for the conditions of single-phase circulation were the part of the investigation. Steady state 1D calculations of the loop were the first step in further developing of calculation method of loop behaviour for single and two-phase circulation at low (atmospheric and subatmospheric) pressures. As it follows from different publications friction factor correlations for isothermal forced flows, especially for laminar regime are not valid for the conditions of buoyancy driven flows, so a modified 1D model has been tried. The modified 1D model was still based on forced flow correlations, but in these correlations the physical properties variability and the existence of thermal and hydrodynamic entrance regions were taken into consideration.

Heat transfer was calculated according to B.S Petuhov’s and A.F. Polyakov’s recommendations for heat transfer and friction for flows with the same direction of forced and natural convection. Besides, the concurrent formation of heat and hydrodynamic boundary layers in the entrance region has been taken into account. In spite of these improvements the calculated flow velocities remained overpredicted in comparison with experimental data.

To study this problem in more details the 2D numerical simulation of velocity and temperature fields in circular tube in model natural circulation loop has been carried out. The present article presents the results for laminar flow.

2. Experimental loop configuration and modified 1D calculation results
The analyzed experimental data were obtained on the natural circulation loop of rectangular configuration the schematic diagram of which is presented in Figure 1 [6, 7].
The flow up (heated) 1 and flow down 3 legs of the loop are joined to the cooling section 2 at the top of the loop. One of the structural features of the loop was large aspect ratio between the down leg cross sectional area and that of the heated leg. This detail of construction made it possible to substantially reduce pressure losses in flow down line (down comer). So the flow characteristics can be considered as somewhat upper limiting case for buoyancy driven flow for given heat flux for specific cross sectional geometry and hydraulic diameter of the heated leg.

Two test sections have been used in the experiments. Both were electrically heated stainless steel circular tubes 1370 mm long and differed by inner diameters (5.4 and 9.1 mm). The test sections can be heated both along their whole length and along the part of it.

The experimental data having been used in present work for comparison with calculated results were obtained for the loop fully filled with the working liquid up to outlet of flow up tube in cooling section. Water was used as working liquid.

Engineering and design methods of predicting thermo and hydraulic characteristics of natural circulation loops are based on 1D description of flow and heat transfer in channels. It is assumed according to this approach that flow velocity and temperature vary only in streamwise direction.

Substantial mathematical simplification of the problem at 1D approach is achieved by introduction of such quantities as heat transfer coefficients and hydraulic loss, which in complex way are connected with the real 3D flow and couldn’t be determined from 1D theory in principle. So one needs to get additional information from experimental data, 3D calculations or semiempirical theories to determine the values of heat transfer coefficients and hydraulic loss.

In present work the modified 1D approach was tested on the experimental data. As it was mentioned above the velocity field pattern in buoyancy driven flows very often is in a transforming state at long distance in streamwise direction. Besides the flow field is forming under the conditions of considerable variety of thermophysical properties of working liquid. So the velocity field structure is highly complex, especially for laminar flows, and developing of the appropriate predictive correlations is not a trivial problem. Furthermore hydrodynamic pattern of natural convection flows is dependent upon channel cross section configuration, channel hydraulic diameter, channel length to hydraulic diameter ratio and flow orientation relatively to gravity acceleration vector. In general case for the above conditions friction factor \( \xi \) must be a function of a number of parameters [8–10]:

\[
\xi = \xi \left( \frac{\rho}{\mu_f}, \text{Re}, \text{Pr}, \frac{\mu_w}{\mu_f}, \text{Gr} \right),
\]

where \( d_h \) – hydraulic diameter, \( \mu_w \) and \( \mu_f \) – dynamic viscosity of liquid at wall and bulk liquid temperature.

According to definition the friction factor is

\[
\tau_w = \frac{\xi}{8} \rho \bar{u}^2 = \mu \left( \frac{\partial \bar{u}}{\partial r} \right)_{r=r_0},
\]

where \( \tau_w \) is wall shear stress, \( \rho \) – liquid density at bulk temperature, \( \bar{u} \) – circulation velocity, \( (\partial \bar{u}/\partial r)_{r=r_0} \) – velocity gradient at the wall.

For the forced flow the effect of physical properties variability on friction factor can be taken into account by applying the correction of the form [8, 9]

\[
\frac{\xi}{\xi_0} = \left( \frac{\mu_w}{\mu_f} \right)^n,
\]
where $\xi_0$ is the friction factor at constant properties, $n > 0$. In general case $n = n(z/d_h, Re, \mu_w/\mu_l)$, that is $n \neq \text{const}$. The most complex $\xi$ dependence on cross-sectional viscosity varying is at laminar flow regimes.

Unfortunately not many quantitative experimental data on friction-viscosity variability dependence and a few predictive correlations are available for today. All of them are classified among forced flows.

In present work the following correlation for $n$ in the relationship (2) for friction factor for flow up section of the loop were used in 1D calculations as the first step [8, 9]:

$$n = C \left( Pe \frac{d_h}{l} \right)^m \left( \frac{\mu_w}{\mu_f} \right)^{-0.062},$$

where $C = 2.30$, $m = -0.3$ at $Pe \cdot (d/h) \leq 1500$ and $C = 0.535$, $m = -0.1$ at $Pe \cdot (d/h) > 1500$. In relationship (3) $Pe$ is Peclet number, $l$ – heated tube length.

In formulas (2) – (3) isothermal flow friction factor $\xi_0$ was taken as $\xi_0 = 64/Re_\nu$, where in the Reynolds number $Re_\nu$ physical properties of the liquid were taken at bulk liquid temperature at current cross-section.

First of all it was necessary to make tests for steady state regimes. In this case the calculation of thermo-hydraulic characteristics of the loop is reduced to solving the balance relationship relative to the circulation velocity

$$\Delta P_{\text{fr}} = \Sigma \Delta P_{\text{loss}},$$

where $\Delta P_{\text{fr}} = \int_L g_1 \rho(l) dl$ – the driving force, $\rho(l)$ liquid density which varies with temperature along the heated zone, $g_1$ – projection of gravity acceleration vector on flow direction $l$, $\Sigma \Delta P_{\text{loss}} = \Delta P_{\text{fr}} + \Sigma \Delta P_{\text{loc}}$ – total hydraulic losses, $\Delta P_{\text{fr}}$ – total hydraulic losses due to friction and $\Sigma \Delta P_{\text{loc}}$ – total hydraulic losses due to local drag reduction. The equation (4) is solved relative to the circulation velocity with the use of iteration procedure.

As the experimental loop [6, 7] was designed in such a way that local drag reduction can be neglected only $\Delta P_{\text{fr}}$ remains in right hand of (4).

For the same directions of forced and natural convection at $q_w=\text{const}$ the local Nusselt number at current cross-section can be determined as [9]

$$\frac{Nu}{Nu_\nu} = \left( 1 + \frac{Gr_\nu}{B Re} \right)^{0.27},$$

where $Nu_\nu$ is Nusselt number for viscous flow with constant physical properties;

$B = 5.40X^4 + 312X^{0.25}$ at $X \leq 0.07$ and $B = 240$ at $X > 0.07$; $X = \frac{1}{Pe \cdot d_h}$; $Gr_\nu = gBd^n q_w/\nu^2 \lambda$ – Grashof number. Equations (2), (3), (5) are modified components in the 1D calculations.

According with the loop design Nusselt number $Nu_\nu$ was calculated with account to practically uniform velocity and temperature distributions in the inlet of the heated section and concurrent formation of heat and hydrodynamic boundary layers in the entrance region [9].

Laminar experimental regimes for the analysis have been chosen according to the condition at the outlet of the heated zone

$$X_{\text{lam}} < X_v \approx 1.3X_v,$$

where $X_v = 12.9(Gr/Re)^{0.8}$ – adjusted coordinate of stability loss of viscous-gravitational flow and $X_v$ is adjusted coordinate of transition to turbulence [9].

Typical example of wall temperature calculations with the above attempts to take into account the variability of physical properties and the differences in temperature and velocity fields as related to fully developed isothermal flow conditions is shown in figure 2. Calculated curves for $d_h = 9.1$ mm for heat flux densities of 3.7 and 17.4 kW/m$^2$ are compared with the experimental data. As it is seen from
In the experiments, it was found that the predicted flow velocities are higher than those actually achieved. This suggests that for low heat fluxes, actual liquid wall shear stresses must be substantially higher than those calculated according to correlations for forced flow. The 2D numerical simulation showed that in low flow regimes, the main effect on wall shear stress is the change of velocity profiles and related to it increase of their gradients at the wall.

For laminar flow, the conservation equations of mass, momentum and energy are those of incompressible Newtonian fluid. They can be written in cylindrical coordinate system as (1)–(3), (5):

\[
\rho \frac{\partial (rU_r)}{\partial r} + \frac{\partial (pU_r)}{\partial z} + \rho U_r \frac{\partial U_r}{\partial r} = 0 \quad (7)
\]

\[
\rho \left( U_z \frac{\partial U_r}{\partial z} + U_r \frac{\partial U_r}{\partial r} + \frac{\partial (\rho U_r)}{\partial r} \right) = - \frac{\partial p}{\partial z} + F_{Bz} + \frac{1}{r} \frac{\partial}{\partial r} \left[ r \mu \left( \frac{\partial U_r}{\partial r} + \frac{\partial U_z}{\partial z} \right) \right] \quad (8)
\]

\[
\rho \left( U_z \frac{\partial U_r}{\partial z} + U_r \frac{\partial U_r}{\partial r} + \frac{\partial (\rho u_r)}{\partial r} \right) = - \frac{\partial p}{\partial r} + \frac{\partial}{\partial z} \left[ \mu \left( \frac{\partial U_r}{\partial r} + \frac{\partial U_z}{\partial z} \right) \right] \quad (9)
\]

\[
\rho \left( U_z \frac{\partial h}{\partial z} + U_r \frac{\partial h}{\partial r} + \frac{\partial (\rho h)}{\partial r} \right) = \frac{1}{r} \frac{\partial}{\partial r} \left[ r \lambda \left( \frac{\partial T}{\partial r} + \frac{\partial T}{\partial z} \right) \right] \quad (10)
\]

Here \( \rho, h, T, p \) are density, enthalpy, temperature and pressure, \( F_{Bz} \) is the projection of buoyancy force vector \( \mathbf{F}_B \) on \( z \)-axis.

The expression for buoyancy force vector, which appears in the system of conservation equations and which is the driving force of motion can be written in general case as \( \mathbf{F}_B(r,z) = \mathbf{g} \left[ \rho(r,z) - \rho_r(z) \right] \), where \( \mathbf{g} \) is vector of gravity acceleration and \( \rho_r \) is local reference density. For inner gravity flows (in closed space bounded by solid walls) the reference density is...
chosen according to the type of inner flow under consideration [11]. In our calculations \( \rho_r(z) \) was considered as cross-section averaged density at current coordinate \( z \) in down leg of the loop.

Boundary conditions were formulated according to the experimental ones in [6, 7]. They are the constant heat flux density on the wall of the heated section and adiabatic conditions on the other surfaces except the cooler zone. The loop cooler was simulated as volumetric heat sink of the cylindrical shape which height was equal to the upper gap in figure 3. The total capacity of the cooler was equal to the input heat power.

The system of conservation equations (7) – (10) was solved numerically with the use of control volume method within the ANES CFD-code [12].

4. 2D simulation results.

The 2D numerical simulation have been carried out for distilled water circulation at heat flux densities which correspond to laminar flow regimes in the experimental loop [6, 7]. The flow was considered to be laminar along the entire length of the heated zone when adjusted coordinate \( X \) at the inlet cross section was less than \( X_{cr} \), the latter was estimated according to relationship (6). The calculation results presented in this paper have been obtained for heating the flow up section with a length of 1300 mm for two tube diameters: 5.4 and 9.1 mm.

The calculated friction factors for two almost twice different tube inner diameters (5.4 and 9.1 mm) and the same heat flux density \( q_w = 2 \text{ kW/m}^2 \) are shown in figures. 4 – 5.

![Graphs showing calculated friction factors and Re(z) numbers for \( q_w = 2 \text{ kW/m}^2 \) and \( d_h = 5.4 \text{ mm} \) as a function of adjusted length.](image1)

4) Calculated friction factors and Re(z) numbers for \( q_w = 2 \text{ kW/m}^2 \) and \( d_h = 5.4 \text{ mm} \) as a function of adjusted length.

The solid lines 3 in the figures 4,a and 5,a are the result of numerical simulation. Dashed-point lines 4 in the same figures correspond to formula (2) for current Re(z) numbers. The longitudinal Re number change with liquid viscosity variation along the heated zone are shown in figures 4,b and 5,b. As the whole the Re number inlet and outlet values under heating conditions can differ from each other from 20 to 40 \% or more depending on heat flux density and hydraulic diameter, the less the diameter the higher the Re number change.

The dashed lines 1 and 2 in figures 4,a and 5,a are friction factors calculated for finite length circular pipe at constant physical properties for inlet and outlet liquid temperatures under heating conditions at given \( q_w \).

As it follows from the figures the friction factor in buoyancy induced flows changes along the tube in a complex way and is highly influenced by the hydraulic diameter. It is also seen that formula (2) noticeably underpredicts longitudinal \( \xi \) distribution.
a) – calculated friction factors as a function of adjusted length for \( d_h = 9.1 \text{ mm} \) and \( q_w = 2 \text{ kW/m}^2 \);  
b) – change of the local Re(\( z \)) number along the tube under heating conditions: (a) 1 – isothermal friction factor at Re = 307 (corresponds to heated zone inlet), 2 – isothermal friction factor at Re = 370 (corresponds to heated zone outlet), 3 – numerical simulation, 4 – formula (2) with account to Re(\( z \)) change under heating conditions.

**Figure 5.** Calculated friction factors and Re(\( z \)) numbers for \( q_w = 2 \text{ kW/m}^2 \) and \( d_h = 9.1 \text{ mm} \) as a function of adjusted length.

In both cases simulated \( \xi \) values always remain higher than that calculated according to formula (2) if the Re number is determined at local bulk liquid temperature. The discrepancy increases downstream and for higher hydraulic diameter one can observe different qualitative behaviour.

Local (at given cross section along the tube) \( \xi \) value is the result of two main effects – viscosity decrease with temperature in near wall region and velocity gradient change due to velocity profile deformation. Velocity profiles calculated for water circulation near the inlet to the heated zone (\( z/d = 3.85 \)) and almost at outlet from it (\( z/d = 139.0 \)) for the tube of \( d_h = 9.1 \text{ mm} \) and two wall heat flux densities \( q_w = 7.0 \text{ and } 20.0 \text{ kW/m}^2 \) are shown in figures 6,a and 6,b.

As it is seen from figures 6,a–b the form of velocity profiles noticeably changes along the heated zone, the extent of velocity profile deformation increasing with the wall heat flux density increase. The velocity profiles deformation is accompanied by the increase of velocity gradient in near wall flow region and the increase of the wall shear stress as a consequence.

More over as it follows from fig.6,c where the longitudinal change of absolute value of wall velocity gradient is presented the velocity gradient increase takes place along the full length of heated tube. That is the evidence of the absence of velocity profile stabilization along the heated zone.

The wall velocity gradient increase is balanced out by the liquid viscosity decrease with temperature in near wall flow region (see figure 6,d). So the result longitudinal wall shear stress behaviour is a function of more strong effect at current flow cross-section. More over the result effect depends not only on wall heat flux density but also on channel cross-section form and hydraulic diameter.

Calculated friction factors as a function of adjusted length for two tubes which differs by inner diameters are shown in figure 7,a. The heating conditions of regimes presented in figure 7 are characterized by close values of the Reynolds number. In the first case (line 1 in figure 7,a) the Re number calculated according to physical properties at local bulk liquid temperature for \( q_w = 15 \text{ kW/m}^2 \) and \( d_h = 5.4 \text{ mm} \) changed from 540 at the inlet to the heated section to 1070 at the outlet from it. In the second case (line 3 in figure 7,a) the corresponding Re number varied for \( q_w = 10 \text{ kW/m}^2 \) and \( d_h = 9.1 \text{ mm} \) from 720 at the inlet to 1040 at the outlet from the heated zone.
a) – b) – velocity profiles at two cross sections of heated tube 9.1 mm inner diameter, calculated for two wall heat flux densities at water circulation; c) – longitudinal change of absolute value of wall velocity gradient corresponding to these regimes; d) – change of liquid viscosity in near wall region due to wall temperature increase;
 a): \( q_w = 7.0 \times 10^3 \text{ kW/m}^2 \), \( 1 - z/d = 3.85 \), \( 2 - z/d = 139.0 \), b): \( q_w = 20.0 \times 10^3 \text{ kW/m}^2 \), \( 1 - z/d = 3.85 \), \( 2 - z/d = 139.0 \), c), d): \( 1 - q_w = 7.0 \times 10^3 \text{ kW/m}^2 \), \( 2 - q_w = 20.0 \times 10^3 \text{ kW/m}^2 \)

Figure 6. Calculated longitudinal change of velocity profiles, liquid viscosity and wall velocity gradient.

One can clearly see from figure 7,a the qualitatively different \( \zeta(\chi) \) behaviour for two diameters. The \( \zeta \) values calculated according to formula (2) are also shown if figure 7,a (lines 2 and 4). It follows from the comparison of curves 1 – 4 that under the conditions of exclusively buoyancy induced circulation wall friction changes along the heated zone in a complex way and friction factor distribution can not be described by a simple relationships of the form \( \zeta = a/Re^b \). In most cases the effect of velocity profiles deformation is predominant as related to the effect of viscosity change. The shape of velocity profiles and the extent of their deformation in its turn depend upon hydraulic diameter.

The shape of velocity profiles for two tube diameters (5.4 mm and 9.1 mm) is shown in figure 7,b as an example. Though qualitatively the shape of both profiles is M-like it is clearly seen from figure 7,b that the deformation of the velocity profiles becomes more intensive with inner tube diameter increase.
5. Conclusions.
With reference to the experimental data obtained for laboratory natural circulation loop a 2D numerical simulation of velocity and temperature fields for laminar low flow regimes have been carried out. It is shown that in natural circulation loop where fluid flow is governed exclusively by buoyancy forces wall shear stresses change along the heated zone in a complex way and

![Graph](image)

a) – calculated friction factors as a function of adjusted length; b) – calculated velocity profiles at the outlet from the heated zone (z=1.28 m); 1 – \( q_w = 15 \text{ kW/m}^2 \), \( d_h = 5.4 \text{ mm} \), numerical simulation, 2 – formula (2) for the regime 1 conditions; 3 – \( q_w = 10 \text{ kW/m}^2 \), \( d_h = 9.1 \text{ mm} \), numerical simulation; 4 – formula (2) for the regime 3 conditions; b): 5 – \( q_w = 10 \text{ kW/m}^2 \), \( d_h = 9.1 \text{ mm} \); 6 – \( q_w = 15 \text{ kW/m}^2 \), \( d_h = 5.4 \text{ mm} \).

**Figure 7.** Calculated friction factors and velocity profiles for different inner diameters of heated tubes.

Friction factor for use in 1D calculations can not be described by simple correlations in the form of \( \zeta = a/Re^b \). The account of the temperature viscous change effect, which though is a substantial factor, does not improve predicting accuracy. In most cases the effect of the velocity profiles deformation, which acts oppositely to viscous decrease with temperature, is predominant. The form of velocity profiles and the extent of their deformation in its turn depend upon the hydraulic diameter. To develop recommendations for 1D loop calculations at low flow regimes one needs to perform parametric calculations with further generalization.

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References
[1] Zvirin Y 1981 A review of natural circulation loops in pressurized water reactors and other systems *Nuclear Engineering and Design* Vol. 67 pp 203 – 225
[2] Kumar N, Nayak A K, Vijayan P K and Vaze K K 2014 Modeling the flow characteristics during start-up of natural circulation systems from rest state *Reactor Engineering Division Research Article* ISSUE NO 336 JAN-FEB 2014 pp 1 – 11
[3] Kumar N, Doshi J B and Vijayan P K 2011 Investigations on the role of mixed convection and wall friction factor in single-phase natural circulation loop dynamics *Annals of Nuclear Energy* V. 38 pp 2247-270
[4] Mousavian S, Misale M, D’Auria F and Salehi M A 2004 Transient and stability analysis in single-phase natural circulation *Annals of Nuclear Energy* Vol. 31 pp 1177-98
[5] Misale M, Garibaldi P, Passos J C and de Bitencourt G G 2007 Experiments in a single-phase natural circulation mini-loop *Experimental Thermal and Fluid Science* Vol. 31 pp 1110-1120
[6] Kaban’kov O N, Zubov N O, Sukomel L A and Yagov V V 2015 Study of flow and heat transfer characteristics under boiling conditions in natural circulation loop Problems in Gasodynamics and Heat and Mass Transfer in Power Plants. Proc. of XX School-seminar of young scientists and specialists under academician A I Leontiev leadership (24 – 29 May 2015 Zvenigorod) (Moscow: MPEI Publishing House) 496 p

[7] Kaban’kov O N, Sukomel L A, Yagov V V and Zubov N O 2016 Unstable circulation regimes during water boiling in a thermosyphon loop under atmospheric pressure Heat Pipe Science and Technology, An Int. Journal (Begel House) Volume 7 Issue 1-2 pp 31-44

[8] Yagov V V 2014 Heat Transfer in Single Phase Media and at Phase Changes (Moscow, MPEI Publishing House) p 542

[9] Petuhov B S, Genin L G, Kovalev S A and Soloviev C L 2003 Heat Transfer in Nuclear Power Plants (Moscow: MPEI Publishing House) p 548

[10] Petukhov B S 1987 Selected Works Heat Transfer Problems (Moscow: Publishing “Nauka”) p 278

[11] Gebhart B, Jaluria Y, Mahajan R L and Sammakia B 1991 Buoyancy-induced Flows and Transport (Moscow: “Mir” Publishing House Translated to Russian) In 2 volumes

[12] Artemov V I and Yankov G G 2010 Numerical analysis of operating efficiency of sectional air-conditioner with air heat exchanger Bulletin of Moscow Power Engineering Institute (Moscow: MPEI Publishing Department) No 6 pp 155–160