Experimental and computational study of a low-pressure natural circulation loop

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Abstract. The experimental and analytical study of single-phase flow and heat transfer in natural circulation loop has been carried out. Experiments were performed on water and ethanol that are the liquids with significantly different thermophysical properties. Experimental apparatus was a rectangular shaped loop with vertical flow up leg. The flow up and flow down legs of the loop are joined to the separator-condenser at the top of the loop. The upper limit of heat flux densities in the experiments was set with the consideration for flow regime to remain in single phase state along the whole heated length. Wall temperature time records being registered at different distances from the inlet to the heated zone indicate the occurrence of temperature fluctuations near the exit from heated zone even at relatively low heat flux densities. This fact displaces a complex changing of velocity profiles along the tube with vortex formation and occurrence of flow instability. Experimental data on longitudinal wall temperature distributions of heated section have been used to test a modified method of hydraulic calculation of the loop. It was pointed out that in spite of long year (since early 1950s) experimental, analytical and numerical investigations of natural circulation loops no suitable predicting recommendations for heat transfer and friction have been proposed till today for engineering hydraulic calculations of single-phase natural circulation loops.

1. Introduction.
Natural circulation heat removal systems find a wide use in various technologies. Many practical applications use single-phase convection of working liquid. In addition, an extended single-phase convection zone can exist in heated section of two-phase circulation systems at low pressures and high liquid subcoolins at the inlet to the heated zone. Passive safety systems in new design nuclear power plants can be taken as an example. 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 thermo gravitational forces is an actual problem. Especially it is important for the analysis of unsteady regimes. 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.

At first glance, it may seem that one will not meet any problems in hydraulic calculations of single-phase natural circulation loop. But this is only at first glance. Liquid flow in natural circulation loop is driven exclusively by the action of thermo gravitational forces, the circulation velocity being set itself
depending on heat power supplied to the heating zone, loop geometry, the presence of local drag reduction elements. That is, the flow of liquid in loop heating zone is nothing else than inner gravity flow in channel. This problem was found not to be closely studied yet in spite of long year (since early 1950s) experimental, analytical and numerical investigations of natural circulation loops. At least no suitable predicting recommendations for heat transfer and friction have been proposed till today for engineering hydraulic calculations of single-phase natural circulation loops.

Numerical simulation [1] have shown that under the conditions of circulation induced exclusively by thermo gravitational forces wall friction changes along the heated zone in a complex way and friction factor distribution can’t be described by a simple relationships of the form $\xi = a/Re^b$.

At present state of the art of loop hydraulic calculation methods forced flow correlations to calculate heat transfer and friction are used in practice. In doing so an important distinctive feature of natural circulation are left aside. These are variability of liquid thermophysical properties and influence of properties variability on friction factor, permanent transformation of the velocity profiles along the heating zone and first of all that circulation itself occurs exclusively as natural convection flow.

Therefore, use in hydraulic calculations of loops the predictive correlations for friction and heat transfer being obtained for the conditions of forced flow with constant thermophysical properties (such an approach one can find in publications very often) unlikely will result to calculated circulation velocity values being in agreement with real circulation velocities in the loop. The analysis [2, 3] have shown that ignoring in hydraulic calculations the above listed distinctive features of natural circulation result to significant overprediction of circulation velocities in comparison with experimental data.

So the developing of engineering method of loop calculations as well as obtaining new experimental data for its verification remains to be an actual problem.

The aim of present work was an experimental and analytical study of single-phase flow and heat transfer in natural circulation loop.

The experiments were carried out on water and ethanol that are the liquids with significantly different thermophysical properties. Experimental data on longitudinal wall temperature distributions of heated section have been used to test a modified method of hydraulic calculation of the loop. The method still uses forced flow correlations to calculate friction and heat transfer but it partially considers the change of flow characteristics due to the affect of mass forces.

2. Experimental apparatus and procedure.

The experimental loop has been designed as a model apparatus to study thermophysical processes at low pressures natural circulation. A schematic diagram of the loop is presented in Figure 1. The flow up (heated) and flow down legs of the loop are joined to the separator-condenser at the top of the loop. One of the structural features of the loop is 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). Cooling heat exchanger and two electrical heaters are installed on the down comer. These members are used for maintaining a specified inlet temperature.

The flow up section was a thin-walled stainless steel circular tube with inner diameter of d=9.1 mm and 146 calibers in length. The flow up leg is consisted of two sections: the upper adiabatic section 54 calibres in length and the lower electrically heated section 92 calibres long.

The wall temperature of the heated part of flow up tube was measured by chromium-alumel thermocouples which hot junctions have been point-welded to the outer surface of the tube. The technique afforded to form thermocouple junctions less than 0.2 mm in dimension. Besides the wall thickness of the heated tube did not exceed 0.45 mm. Thus the thermal response time of the test section was rather small and one could record temperature fluctuations (which in some regimes were very intensive).
Figure 1. A schematic diagram of the experimental loop. 1 - heated section; 2 – adiabatic section; 3 – separator-condenser; 4 – down comer.

Temperature recordings were realized with data acquisition system, which affords high-frequency sampling of measured quantities. The frequency of thermocouples sample was 100 Hz. It was enough to obtain practically simultaneous temperature records for the whole heated length.

Present data have been obtained for the loop fully filled with the working liquid up to outlet of flow up tube section in separator.

The upper limit of heat flux densities in the experiments was set with the consideration for flow regime to remain in single phase state along the whole heated length. For example, when liquid temperature at the inlet to the heated section was set within the range of (20÷30) °C the upper limit values of heat flux densities where 25 kW/m² for water and 10 kW/m² for ethanol

3. Experimental results.

Typical wall temperature time records at different distances from the inlet to the heated zone for circulation regimes with closely spaced inlet liquid temperatures and different heat flux densities are shown in Figure 2 (water) and Figure 3 (ethanol).

As it is seen from Figure 2 and Figure 3, one can observe wall temperature fluctuations near the exit from heated zone even at relatively low heat flux densities. As heat flux density increases the fluctuation amplitude increases and the start section of fluctuation regime displaces towards the inlet to the heated section. This fact displaces a complex changing of velocity profiles along the tube with vortex formation and occurrence of flow instability. In this case, the colder liquid portions from the flow core stream towards the wall, where their mixing with more heated near wall liquid layers takes place.

According to [4 – 6], the critical adjusted coordinate $X_*$ of viscous-gravitational flow stability loss is

$$X_* = 1 \left( \frac{z}{d} \right)_{*} = 12.9 \left( \frac{Gr}{Re} \right)^{0.8}$$

Substitution of the experimental $Gr$ and $Re$ numbers values into (1) gives one $(z/d)_*$ values, which show an agreement with the experimental $z$ coordinates of wall temperature fluctuations start within the error range of 30 %.
Figure 2. Heated tube wall temperature time records at single-phase convection of water in the loop

Figure 4 and Figure 5 show the examples of experimental wall temperature distributions along the heated section for water and ethanol respectively. The points on the graphs are the result of averaging the thermocouple readings.

One sees qualitatively typical curves for the case of single-phase convection without any salient features. Here a natural question arises: to what extent the proposed in literature hydraulic calculation methods of single-phase natural circulation loop agree with the experimental data obtained.

As already have been noted, the hydraulic calculation of a single-phase natural circulation loop is not a trivial task, since the question of the correct method for calculating the friction factor resistance and heat transfer coefficient still remains open.

A modified hydraulic calculation method of the single-phase natural circulation loop, which at to-day state of the art seems to be the most correct has been tested on experimental data in present work. The method is also based on forced flow correlations but it takes into account the substantial effects in flows under the influence of thermo gravitational forces. The description is given below.
Figure 3. Heated tube wall temperature time records at single-phase ethanol convection in the loop

Figure 4. Experimental longitudinal wall temperature distributions of heated section at water flow

Figure 5. Experimental longitudinal wall temperature distributions of heated section at ethanol flow

4. Modified hydraulic calculation method
When flow of liquid is driven exclusively by thermo gravitational forces, the velocity field pattern is in a transforming state in streamwise direction. Besides the flow field is forming under the conditions of considerable variety of thermophysical properties of working liquid.

In general case 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 and friction factor \( \xi \) must be a function of a number of parameters \([4-6]\): \( \xi = \xi(z/d_h, \text{Re}, \text{Pr}, \mu_w/\mu_f, \text{Gr}) \), where \( d_h \) – hydraulic diameter, \( \mu_w \) and \( \mu_f \), – dynamic viscosity of liquid at wall and bulk liquid temperatures.

Therefore even if the absence of well-grounded recommendation for calculations flows in tubes exclusively driven by thermo gravitational forces makes one to use in 1D calculations correlations for forced flow it must be at least taken into account such obvious factors as temperature variety of thermo-physical properties of liquid (viscosity first of all) and its influence on friction factor and heat transfer coefficients change under the additional effect of natural convection on forced flow \([4-6]\).

In addition, a details of loop construction must be taken into account. As for laboratory loop, which was used in present study it should be considered in heat transfer and friction calculations the simultaneous formation of hydrodynamic and thermal boundary layers along the heated zone.

Present calculations are made 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

\[
\Delta P_{df} = \Sigma \Delta P_{loss} ,
\]

where \( \Delta P_{df} = \oint g \rho(l) dl \) – the driving force, \( \rho(l) \) liquid density which varies with temperature along the heated zone, \( g \) – projection of gravity acceleration vector on flow direction \( l \), \( \Sigma \Delta P_{loss} = \Delta P_{f} + \Sigma \Delta P_{loc} \) – total hydraulic losses, \( \Delta P_{f} \) – total hydraulic losses due to friction and \( \Sigma \Delta P_{loc} \) – total hydraulic losses due to local drag reduction.

The aim of the calculation is to determine the circulation velocity \( w_0 \) at which eq. (2) 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 \( T_{in} \). The equation (2) is solved relative to the circulation velocity with the use of iteration procedure.

The main difficulty involves the choice of the correct correlations to calculate friction factors and heat transfer coefficients in streamwise direction.

The modified hydraulic calculation method to be verified with experimental data uses the correlations given below.

**Transition from laminar to turbulent flow.** Under the influence of thermo-gravitational forces on forced laminar flow the stability loss of such flow and transition to turbulence occurs at lesser Re numbers. For example, for circular tube the above transition takes place at \( \text{Re} = \text{Re}_c < 2300 \).

The adjusted length, at which the viscous-gravitational flow loses stability, that is the length of stability loss \( X_c \) are estimated by the empirical correlation (1) \([5]\). The transition to turbulent flow takes place at somewhat greater \( X \) value \([5]\):

\[
X = X_c \approx 1.3 X^* .
\]

If adjusted length value \( X = (1/\text{Re})(z/d) \) for current cross section of the flow is less than \( X_c \) value for the same cross section flow is considered to be laminar, else flow is considered to be turbulent.

**Laminar flow.** Search a sufficient correlation to calculate friction factor for laminar flow regime is the most difficult task. Velocity field pattern in this regime changes along the flow direction in a most complex way.

Velocity field is in permanent transformation mode under the conditions of working liquid property variability. In addition, 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 \([4-6]\):
\( \xi = \xi(z/d_h, \text{Re}, \text{Pr}, \mu_w/\mu_f, \text{Gr}) \), where \( d_h \) – hydraulic diameter, \( \mu_w \) and \( \mu_f \) – dynamic viscosity of liquid at wall and bulk liquid temperature. 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.

The effect of physical properties variability on friction factor in present calculation method is taken into account by applying the correction of the form \[4, 5\]

\[
\frac{\xi}{\xi_0} = \left( \frac{\mu_w}{\mu_f} \right)^n
\]

(4)

where \( \xi_0 \) is the friction factor at constant properties, \( n > 0 \). In general case \( n = n(z/d_h, \text{Re}, \mu_w/\mu_f) \), that is \( n \neq \text{const} \).

In present calculation method the following correlation for \( n \) in the relationship (4) for friction factor for flow up section of the loop were used in 1D calculations as the first step \[4, 5\]:

\[
C = 0.62.
\]

(5)

where \( \xi_0 = (1.82 \log \text{Re} - 1.64)^{-2} \).

The Nusselt number in the loop is \( \text{Nu}_{\text{loop}} = \text{Nu}_v C_{\text{vis}} C_{\text{nc}} \).

Here \( C_{\text{vis}} \) is the correction factor for the viscosity variability influence, \( C_{\text{nc}} \) is the correction factor for the effect of natural convection on forced flow.

Influence of viscosity variability is accounted as \( C_{\text{vis}} = (\mu_w/\mu_f)^n, n = -1/6 \) at \( q_c = \text{const} \), \( \text{Nu}_v \) is the Nusselt number for viscous flow with constant physical properties.

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 \[5\]

\[
\text{Nu}_v /\text{Nu}_{v,0} = 0.35[(1/\text{Pe})(\text{d}/\text{d})]^{-1/6}[1+2.85[(1/\text{Re})(\text{d}/\text{d})]^{0.42}]
\]

where \( B = 5.40X + 312X^{0.25} \) at \( X \leq 0.07 \) and \( B = 240 \) at \( X > 0.07 \); \( X = \frac{1}{\text{Pe} d_h} ; \text{Gr}_q = g/\text{d}_h^4 q_w/(\rho^2 \lambda) \) – Grashof number.

According with the loop design Nusselt number \( \text{Nu}_0 \) 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 \[4, 5\].

\[
\text{Nu}_v /\text{Nu}_{v,0} = 4.36 \text{ at } q_w = \text{const}.
\]

Turbulent flow. Influence of viscosity variability on friction factor is accounted as

\[
\frac{\xi}{\xi_0} = 1 - (1/2)(1+M)^{n} \log(1+M),
\]

where \( M = (\mu_w/\mu_0 - 1)(\mu_w/\mu_f)^{0.17}; n = 0.17 - 2 \cdot 10^{-6} \text{Re} + 1800 \text{Re} \); \( \xi_0 = (1.82 \log \text{Re} - 1.64)^2 \).

The Nusselt number in the loop is \( \text{Nu}_{\text{loop}} = \text{Nu}_0 C_{\text{vis}}, \) where \( C_{\text{vis}} \) is the correction factor for the viscosity variability influence \( C_n = \text{Nu}/\text{Nu}_0 = (\mu_w/\mu_f)^n, n = -0.11 \) for heating conditions.
$$\text{Nu}_0 = \frac{(\xi/8) \text{Re Pr}}{1 + \frac{900}{\text{Re}} + 12.7 \left(\frac{\xi}{8}\right) \left(\text{Pr}^{2/3} - 1\right)}$$

5. Comparison with experimental data.
Calculated and measured longitudinal wall temperature distributions are shown in Figure 6 (water) and Figure 7 (ethanol).

For low heat flux densities, the 1-D calculations predict wall temperature values lower than experimental ones (that is the calculated circulation velocity is appreciably higher than that which actually was reached in the experiments). As heat flux densities increase the discrepancy between calculated and experimental longitudinal wall temperature distributions decreases. As has been shown in[2, 3] in mixed circulation regimes when the boiling onset takes place near the outlet from the heated zone there is a good agreement between calculations and experiments. So, when circulation velocities are higher than some low boundary velocity the single-phase loop flow, which is governed exclusively by thermal gravity forces begins to acquire the properties of forced flow with predominating influence of thermal gravity. At lower circulation velocities (lower than the above boundary velocity limit) as it follows from the comparison between calculated and experimental wall temperature distributions for

Figure 6. Measured and calculated longitudinal wall temperature distribution of the heated wall for water flow (1 – experiment, 2 – calculations)
low heat flux densities actual liquid wall shear stresses must be substantially higher than that calculated according to correlation for forced flow even with the account of the correction for viscous variability.

At low heat flux densities laminar flow regime existed most likely in the greater part of heated tube according to observation of the wall temperature time records. In this flow regime velocity field and correspondingly friction factor changes in most complex way. As it follows from the comparing experimental and calculated curves for low heat flux densities a single correction for viscous variability in the form of (4)–(5) is not enough to get a correct calculation formula for friction factor.

\[ q = 3.4 \text{ kW/m}^2, \quad T_{\text{in}} = 28.0 \, ^\circ\text{C} \]
\[ q = 5.0 \text{ kW/m}^2, \quad T_{\text{in}} = 30.0 \, ^\circ\text{C} \]
\[ q = 9.6 \text{ kW/m}^2, \quad T_{\text{in}} = 19.0 \, ^\circ\text{C} \]

**Figure 7.** Measured and calculated longitudinal wall temperature distribution of the heated wall for ethanol flow (1 – experiment, 2 – calculations, 3 – saturation temperature)

### 6. Conclusions

New experimental data on flow and heat transfer characteristics for water and ethanol have been obtained in natural circulation loop for single-phase flow regimes.

Experimentally obtained longitudinal wall temperature distributions in heated zone have been used to verify a modified method for hydraulic calculation of single-phase natural circulation loops. Method uses forced flow correlations to calculate friction and heat transfer but it partially considers the change of flow characteristics due to the affect of mass forces, in particular a corrections for viscous variability influence on friction factor and heat transfer coefficient are taken into account.

At low wall heat flux densities, the calculated values of wall temperature are noticeably lower than measured ones. This indicates that the calculated circulation velocities are higher than those, which actually have been set in the loop. As the heat flow increases, the discrepancy calculated and measured values decreases, and under conditions close to boiling of the liquid, the calculated temperature curves coincide with the measured ones.

As for low heat flux densities a single correction for viscous variability in the form of (4)–(5) is not enough to get a correct calculation formula for friction factor.

### Acknowledgments

The work was supported by the RFBR grant No. 19-08-01044

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