Analysis of methods for calculating intra-channel boiling heat transfer in refrigerants

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Abstract. The paper analyzes and compares methods for calculating heat transfer during boiling of refrigerants in horizontal pipes, as well as works devoted to the study of two-phase flows in minichannels. The main provisions of the comprehensive analysis method based on true phase parameters were formulated. It was found that for all the values of the mass velocity, there is a significant discrepancy in the calculated data, while the method based on the comprehensive analysis provides the best agreement with the experiment. The comprehensive method was tested in relation to the analysis of heat-hydrodynamic processes at boiling in minichannels. Comparison of the calculation results using the method based on true phase parameters with the experimental data showed a good agreement.

1. Introduction
Boiling of liquids in pipes and channels is used in many designs of heat exchange equipment in power engineering, refrigeration, food and chemical technologies. Due to the complexity of the boiling process, heat-hydrodynamic problems in these cases are usually not solved analytically, but empirical and semi-empirical approaches are often contradictory and not reliable enough.

The problem becomes even more urgent in connection with the development of "breakthrough" technologies for heat transfer intensification in heat exchange apparatus construction and the appearance of new designs with boiling in slits and minichannels. In all works devoted to the problem of intensification, an increase in heat transfer coefficients by 40% or more is stated, but the issues of heat-hydrodynamic calculation remain open.

2. Purpose of the study
The purpose of the study is to test a comprehensive method for analyzing heat transfer during in-pipe boiling, based on true phase parameters, for boiling in pipes and minichannels.

The research objectives are:
- comparison and analysis of known methods for calculating heat transfer during in-pipe boiling of refrigerants;
comparison of experimental data on heat transfer during boiling in pipes and minichannels with calculation using a comprehensive method.

3. Research Methods

To calculate the heat transfer during the boiling of liquids in pipes, in most cases, dependencies are used to calculate the average heat transfer coefficients along the length of the pipe ("average heat transfer formulas"). In this case, as a rule, the given phase parameters are used, calculated from the equations of the material and heat balances.

The main consumption parameters are as follows:

Mass vapor content:

\[ X = \frac{M''}{M' + M''}, \]  

where, \( M \) – total mass flow rate of the mixture; \( M'' = M \cdot X \) – mass flow rate of vapor; \( M' = M \cdot (1 - X) \) – mass flow rate of the liquid.

Volume consumption vapor content:

\[ \beta = \frac{1}{\left[ 1 + \frac{(1 - X) \rho''}{\rho'} \right]^2}. \]  

Mass rate:

\[ w' = \frac{M}{\rho \cdot \rho'}. \]  

Circulation rate:

\[ w_0 = \frac{M}{\rho \cdot \rho'}. \]  

Superficial vapor velocity:

\[ w'' = \frac{w \cdot \rho}{\rho'}. \]  

Superficial liquid velocity:

\[ w' = \frac{w \cdot (1 - x)}{\rho'} = w_0 \cdot (1 - x). \]  

Mixture velocity:

\[ w_{mix} = w'' + w'. \]  

A widely used formula for calculating average heat transfer is an equation of the form [1]:

\[ \alpha = A \cdot q^{0.6} \cdot (w \cdot \rho)^{0.2} \cdot d_{out}^{-0.2}. \]  

As in most methods of this type, the effect of \( q^{0.6} \) bubble boiling and \( w \cdot \rho^{0.2} \) convection are expressed in constant values. This formalization of the process does not take into account the change in the heat transfer mechanism associated with a change in the ratio of velocity and heat flow.

From this point of view, the approach of S. S. Kutateladze is interesting, which provides for a variable influence of \( \alpha_0 \) bubble boiling and \( \alpha_{con} \) convection.

\[ \alpha = \alpha_{con} \cdot \left[ 1 + \left( \frac{\alpha_0}{\alpha_{con}} \right)^2 \right]. \]  

One of the promising directions for the development of heat-hydrodynamic analysis of two-phase flows in a confined space is an integrated approach based on the interrelated analysis of the following characteristics [2]:

– true phase parameters (phase slip);
– two-phase flow modes;
– local heat transfer;
– pressure losses during the two-phase mixture flow.

The main true parameters are:

True volume vapor content:

\[ \varphi = \frac{f''}{f_{sec}}, \]  

(10)

where, \( f'' \) – a part of the pipe's section filled with vapor, \( f_{sec} \) – the total cross section of the pipe.

True liquid velocity:

\[ w' = \frac{w_0'}{(1-\varphi)}, \]  

(11)

True vapor velocity:

\[ w'' = \frac{w_0''}{\varphi}. \]  

(12)

Slip coefficient:

\[ S = \frac{w''}{w'}. \]  

(13)

Finding the true parameters is a separate rather complex thermophysical problem. Methods for calculating the true parameters for boiling in pipes with a diameter of more than 6 mm are presented in [2].

In particular, for refrigerants that are boiling in pipes, it is recommended that:

\[ \beta - \varphi = 0.06 \cdot \beta \cdot (1 - \beta)^{0.5} \cdot \left(\frac{Fr_0}{Re_0}\right)^{-0.23} \cdot \left(\frac{P_0}{P_{cr}}\right)^{-0.15}, \]  

(14)

where, \( \beta \) – volume consumption vapor content; the \( Fr_0 \) and \( Re_0 \) criteria are calculated by circulation rate (4).

Using a comprehensive approach, an equation was obtained for calculating the heat transfer coefficients for wave and stratified modes when boiling R12 and R22 in pipes [2]

\[ \alpha = 0.695 \cdot (w \cdot \rho)^{0.85} \cdot q^{0.047} \cdot \left(\frac{P_0}{P_{cr}}\right)^{-1.1}, \]  

(15)

The flow modes map [3] and the dependence (14) were used for calculating the true volume vapor content.

To calculate the local heat transfer in the ring mode, the following equation was obtained:

\[ \alpha_{ring} = \alpha' \left[ 1 + 36.5 \cdot 10^{-9} \left(\frac{w'' \cdot r \cdot q'}{q} \right)^{1.5} \cdot \left(\frac{\alpha_{b.o}}{\alpha'}\right)^2 \right]^{1/2}, \]  

(16)

where, \( \alpha' \) – the convective component of single-phase liquid convection, \( \alpha_{b.o.} \) – the bubble boiling component [2].

The comparison of the data obtained by means of equations for calculating the average heat transfer and the calculations made with the help of the comprehensive method using true phase parameters is of considerable interest.
4. Results and discussion

Figure 1 shows the result of comparing calculations using equations (8), (9), (15) with experimental data from H. Yusida and S. Yamoguchi [4].

The presented data make it possible to conclude that at the mass velocity \( w\rho = 50 \text{ kg/(s} \cdot \text{m}^2) \) at \( q < 3000 \text{ W/m}^2 \), the differences in the calculated values of \( \alpha \) obtained from the formulas for the average heat transfer coefficients are very significant. The explanation for this is that each of the equations was obtained for the boiling conditions in pipes \( d_0 > 15 \div 20 \text{ mm} \) and do not take into account the specifics of the flow in the pipe \( d_0 = 6\text{ mm} \). At the same time, as the diameter decreases, the difference between true and consumption parameters increases, so the use of the latter leads to significant errors.

At \( w\rho = 50 \text{ kg/(s} \cdot \text{m}^2) \) and the values of the mass consumption vapor content \( X = 0.1 \div 0.8 \), a wave flow mode was detected using the mode diagram [3]. Using equation (15), local values of heat transfer coefficients averaged in the range \( X \) (line 3) were calculated.

The maximum deviations of the values obtained by the formula (15) from the calculation by the equations for the average heat transfer coefficients at \( q > 2 \text{ kW/m}^2 \) are within 30%. At the same time, the types of dependencies differ qualitatively, which is explained by the difference between true and consumption parameters.

According to the mode map, at \( w\rho = 150 \text{ kg/(s} \cdot \text{m}^2) \) under experimental conditions, the ring flow mode is observed. The calculation using equation (15) (line 4) gives a good agreement with the experimental data [4]. The calculation results for equations (7) and (8) are 45–55% lower than the experimental data. This situation is explained by the following. Firstly, the formulas for average heat transfer coefficients do not take into account flow modes and were obtained mainly at mass velocities \( w\rho = 50 \div 100 \text{ kg/(s} \cdot \text{m}^2) \) in pipes with a diameter \( d_0 = 10 \div 20 \text{ mm} \). For these conditions, the ring mode, which is characterized by the highest values of heat transfer coefficients, is usually not achieved. In addition, as in the previous case, the difference between the true parameters (equation (15) and the expenditure parameters (equations (8) and (9)) is significant, the latter parameters, under certain conditions, have a limited physical meaning.
Approbation of the comprehensive method based on true parameters and developed for pipes in relation to boiling in minichannels is of particular interest.

Minichannel technologies with phase transitions of working substances are currently a very promising direction of heat exchange apparatus construction [5–8]. Minichannels began to be widely used in power engineering, chemical technology, computer systems, and compact air conditioning systems. Minichannel technologies in refrigeration are of highly relevant today.

According to [7], the heat transfer coefficient in minichannels is 2.5 times higher than in plate devices, and the amount of working substance that provides similar performance is up to 4 times lower. In comparison with plate heat exchangers with minichannels, they have improved weight and size characteristics, have increased strength and manufacturability.

However, the calculation of the heat-hydrodynamic characteristics of evaporators with boiling in minichannels is an open task of thermophysics.

The minichannel flow mode map was used for analysis [8].

The unified diagram of two-phase flows in a confined space.

The unified diagram includes two ranges of Froude criteria:
- \( Fr_{sm} < 10^5 \) – pipe operating range \( d_0 = 6 \div 20 \) mm;
- \( 10^5 < Fr_{sm} < 10^7 \) – minichannel operating range \( d_h = 0.5 \div 1.6 \) mm.

The diagram unifies:
- experimental data on the pipes with the R12, R22, R134a, R717 refrigerants, water at \( \rho w = 50 \div 250 \) kg/(s\( \cdot \)m\(^2\)) \( t_0 = \) from +20 to –20°C
- experimental data on the minichannels with the R134a, R410a refrigerants, at \( \rho w = 110 \div 650 \) kg/(s\( \cdot \)m\(^2\)), \( t_0 = \) from –10°C to +20°C.
For comparison, figure 3 shows experimental data on the true volume vapor content when boiling R12 in a pipe with a diameter of 6 mm at a boiling temperature of –10°C and for refrigerant R134a boiling in a minichannel with an equivalent diameter of \(d_h = 0.5\) mm.

At the same speeds, the values of \(\varphi\) during boiling in the pipe are 6 ÷ 12% higher than in the minichannel, although they are qualitatively consistent with each other.

From the above, it follows that the difference between the true volume vapor content and the flow rate is greater in the minichannel than in the pipe. This indicates a greater difference between the true and superficial phase velocities in minichannels compared to pipes. Therefore, the use of the comprehensive method of analysis is most relevant for minichannels.

In minichannels, the difference between true and consumption parameters decreases with increasing speed. Figure 3 shows that the values of \(\varphi\) approach \(\beta\) as the speed increases. At mass velocities of \(w_p > 500\) kg/(s·m²) and at values of mass consumption vapor content \(X > 0.6\), it is legitimate to use a homogeneous model (\(\varphi = \beta\)) using the consumption parameters.

In relation to the flow in minichannels, the equation given below provides a sufficient degree of accuracy (17).

\[
\varphi = \beta - 0.6 \cdot \beta \cdot (1 - \beta)^{0.5} \cdot \left(\frac{\sigma \cdot We}{\mu \cdot d_h^2 \cdot \rho_f \cdot \rho_i} \right) \cdot \left(\frac{P_e}{P_{cr}}\right).
\]

(17)

where, Weber’s criterion: \(We = \frac{\rho_e \cdot D_h \cdot w_{in}}{\sigma}\).
Based on the comprehensive method using the mode map (figure 2), the equation for calculating the true vapor content (17) and the equation (16) for local heat transfer in the ring mode, the heat transfer coefficients for boiling of R134a in the minichannel were calculated for the conditions [7].

\[
\text{Heat transfer coefficient, } k = \frac{W}{m^2 \cdot K}
\]

\[
\begin{align*}
1 - G &= 632.5 \text{ kg/(m}^2\text{ s)} \\
4 - G &= 464.3 \text{ kg/(m}^2\text{ s)} \\
7 - G &= 214.2 \text{ kg/(m}^2\text{ s)} \\
10 - G &= 153.3 \text{ kg/(m}^2\text{ s)} \\
13 - G &= 155.0 \text{ kg/(m}^2\text{ s)} \\
2 - G &= 567.5 \text{ kg/(m}^2\text{ s)} \\
5 - G &= 416.0 \text{ kg/(m}^2\text{ s)} \\
8 - G &= 303.8 \text{ kg/(m}^2\text{ s)} \\
11 - G &= 105.4 \text{ kg/(m}^2\text{ s)} \\
12 - G &= 630.0 \text{ kg/(m}^2\text{ s)} \\
3 - G &= 525.7 \text{ kg/(m}^2\text{ s)} \\
6 - G &= 365.5 \text{ kg/(m}^2\text{ s)} \\
9 - G &= 253.4 \text{ kg/(m}^2\text{ s)} \\
14 - G &= 365.5 \text{ kg/(m}^2\text{ s)} \\
15 - G &= 630.0 \text{ kg/(m}^2\text{ s)}
\end{align*}
\]

Figure 4. The comparison of experimental data [7] with the calculation according to equations (16) and (17) for: R134a, \( t^0 = +30^\circ \text{C} \), \( dh = 0.5 \text{ mm} \).

The comparison results show that in the range of \( x = 0.04 \div 0.4 \), the calculation using equations for local heat transfer with true velocity values describes experimental data on boiling in the minichannel with an average accuracy of 10%, both at mass velocities \( G = 630 \text{ kg/(m}^2\text{ s)} \) and at \( G = 155 \text{ kg/(s} \cdot \text{m}^2\text{)} \).

5. Conclusion

1. When making the calculation of heat transfer during the boiling of refrigerants in small-diameter pipes, the best result is provided by the dependence of local heat transfer for the ring mode using the true phase parameters.

2. The developed method for calculating local heat transfer during in-pipe boiling, based on the true flow parameters, describes experimental data on boiling in minichannels with satisfactory accuracy.

3. The development of a comprehensive method for studying heat transfer and pressure losses using true parameters in the flow area in minichannels at negative temperatures, as well as on surfaces with intensified coatings and turbulators is a promising direction for further research.

4. The optimization of the heat-hydrodynamic parameters is an important research direction.

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