HEAT TRANSFER IN VAPOUR-LIQUID FLOW OF CARBON DIOXIDE

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ABSTRACT

During the last decade a number of studies of boiling heat transfer in carbon dioxide notably increase. As a field of CO_2 practical using corresponds to high reduced pressures, and a majority of available experimental data on CO_2 flow boiling even in submillimetric channels relate to turbulent liquid flow regimes, a possibility arises to develop sufficiently general method for HTC predicting. Under the above conditions nucleate boiling occurs up to rather high flow quality, even in annular flow regime due to extremely small size of an equilibrium vapour bubble. This conclusion is in agreement with the available experimental data. The predicting equation for nucleate boiling heat transfer developed by the present author in 1988 is valid for any nonmetallic liquid. A contribution of forced convection in heat transfer is calculated according to the Petukhov et al. equation with correction factor, which accounted for an effect of velocity increase due to evaporation. This effect can be essential at relatively small heat fluxes and rather high mass flow rates. The Reynolds analogy and homogeneous model are used in order to account for the convective heat transfer augmentation in two-phase flow. Due to low ratio of liquid and vapour densities at high reduced pressures the homogeneous approximation of two-phase flow seems to be warranted. A total heat transfer coefficient is calculated as an interpolated value of boiling and convective HTCs. The experimental data on CO_2 flow boiling related to regimes before heated wall dryout incipience are in rather good agreement with the calculations.

INTRODUCTION

Beginning from 1990s carbon dioxide is considered as a perspective alternative refrigerant. Bearing in mind that this substance was widely used for ship refrigerators in the end of the 19th century, one can say about a reappraisal and a return of the old refrigerant [1]. The main trait of carbon dioxide as a refrigerant is rather high critical pressure (p_c=7.38MPa), low critical temperature (T_c=304.1K) and high pressure at the triple point (p_{tp}=0.518MPa), the reduced pressure at the triple point, p_{tp}/p_c being 0.07. It means that at the temperature range typical for refrigerating systems its saturation pressure is much higher than for traditional refrigerants. For example, for carbon dioxide the saturation temperature range: -10 – 0˚C corresponds to pressures from 2.65 up to 3.48 MPa, the reduced pressures being approximately 0.35-0.5. High working pressures present now a limiting factor for wider use of carbon dioxide in industrial systems, but this can not be a long-term one: compressors, pumps, valves are already on the market suited to operation at 4 MPa [1]. Already now this refrigerant is used in a number of applications, including mobile air conditioning, residential heat pump and hot water heat pump systems [2]. Very interesting idea to use carbon dioxide in atomic energetics is discussed in [3]. The authors of this paper consider nuclear cogeneration systems for LWRs and HTGRs (light water and high temperature gas reactors correspondingly). In particular, in LWR it is assumed to recover all waste heat by means of using boiling heat transfer of liquid carbon dioxide at 20°C instead of water as a cooling medium; the heat is then utilized for local heat supply. The CO_2 gasified in the cooling process is used as a working fluid of mechanical heat pumps for hot water supply. The thermodynamic cycle as is clear is projected for high reduced pressure; the high level pressure has to be supercritical one.

Due to high reduced pressures, which are typical for carbon dioxide using, HTCs (heat transfer coefficients) in boiling obtained in experimental studies are usually much higher than for traditional refrigerants at the same saturation temperature [4]. This advantage of carbon dioxide can not be considered as a consequence of its particular thermophysical properties, it originates from the higher level of reduced pressures in CO_2 boiling. In the review paper [5] it is shown that main regularities of CO_2 pool boiling are similar to the ones for other liquids at the close reduced pressures. The authors of the later paper note that the most part of boiling heat transfer experiments have been conducted under forced flow conditions in channels. Few available experimental results on pool boiling of carbon dioxide including the works of the authors of the review paper [5] show that HTCs are slightly higher than it follows from the empirical calculation method suggested in the widely known German Heat Atlas.

Thome and Ribatski [4] conducted an overall review of flow boiling heat transfer and two-phase flow of CO_2 in the literature and found that not one of the available prediction methods was able to predict the experimental data of CO_2 well. The authors of [4, 6] assume that the problem is in some specific features of carbon dioxide, which have to be reflected in new predicting correlations for carbon dioxide boiling. Prof. J. Thome several years ago has advanced a just opinion that knowledge on flow pattern plays for two phase heat transfer prediction a similar role as information on flow regime in single-phase heat transfer. His scientific team during the last decade has suggested some models for determination of two phase flow pattern, mainly for horizontal channels. On this basis they derived the predicting correlations for heat transfer at the corresponding conditions. In the last year publications [2, 6] a new prediction method for flow boiling heat transfer especially for carbon dioxide using their modified flow patterns map has been presented. The method appears to be rather successful, as 71.4% of the experimental points are in the band of ±30% from the calculated values of HTC. Unfortunately, the developed prediction method can not be considered as a general one, because the majority of the correlations including the equation for HTC in nucleate boiling involve some empirical parameters in order to fit the data on carbon dioxide flow boiling. Certainly, the present author is fully aware great difficulties in modelling any actual process of momentum and energy transfer in two-phase flow; in this relation even a combination of physical models with empirical information can be useful not only for practical calculations, but also for better understanding of the phenomenon mechanism. However, development of general and physically grounded models maintains more attractive aim.

It is obvious enough that the main reason of deviation of some experimental data on two-phase flow and heat transfer from prediction is an absence of strict theory of the process, but not the specific properties of one substance or another. In reference to nucleate boiling heat transfer this issue has been considered in detail in [7, 8]. Due to principal impossibility to build up a closed mathematical description of nucleate boiling heat transfer there are two main ways in order to obtain the predicting equations for HTC. The first one is formation of new and new empirical correlations to account for specific features of a substance, of a heated wall, and of reduced pressures range. This way seems to be a dead-end one. The second way is development of approximate theory of the process, which allows deriving a predicting equation for some typical ("average") conditions with accuracy to few numerical factors. Such approach has been realized for nucleate boiling heat transfer by the present authors 20 years ago [9], two numerical factors being fitted to the experimental data. Certainly, the theory can not account for all specific features of a particular boiling regime, first of all, wall roughness characteristics, but the main quantitative regularities of the process are based theoretically and rather strictly. In principle, under some particular conditions there are no difficulties to introduce in the equation (in the values of two numerical constants in it) small corrections in order to increase its predicting capacity, these corrections being valid for the definite combination of a substance, wall, and pressure range.

This approach is suggested to apply to the discussed problem of flow boiling heat transfer of carbon dioxide. Although, as is known, at developed nucleate boiling heat transfer intensity practically does not depend on flow velocity, in the discussed case of carbon dioxide forced flow boiling some features of available experimental data must be taken into account. The high reduced pressure means a low vapour specific volume that allows using small diameter channels and compact heat exchangers. Naturally, the majority of the experimental data on carbon dioxide flow boiling discussed in the review paper [4] and partly in the review paper [5] have been obtained in the channels of small cross-section. Besides, the distinction of the available experimental data is relatively small heat fluxes and maintaining high intensity of heat transfer up to high flow qualities. It is obvious that flow patterns can be very different at the greatly different flow qualities, but HTC keeps practically the same value.

Fully recognizing the principal idea of the Swiss colleagues on an importance of flow pattern for heat transfer in two-phase flow it is necessary to say that some particular results of their analysis [2] seem to be doubtful. First of all, the authors factually used as a basis the Taitel and Dukler approach [10], which considers the stratified flow with smooth horizontal interface as an initial state for analysis. Certainly, this approach developed later for channels of any orientation [11] has doubtless advantages in comparison to many other flow pattern maps, as it is more grounded on simple but clear physical models. In particular, an idealized scheme with the smooth horizontal interface can be approximately accepted for large diameter channels. But it can not be applied to the channels with diameter of an order of the Laplace constant. It is impossible to agree with the authors of [2] when they estimate gas and liquid slip velocity in horizontal channels introducing effect of gravitational rise of a bubble or use the Kutateladze formula for CHF in pool boiling in order to find conditions of wall dryout. One can expect that in the case of carbon dioxide flow boiling a possibility exists to use more general and simpler approach to analyze two-phase flow pattern. The reason is high reduced pressures, which correspond to the range of liquid phase existence of CO_2.

This paper gives an analysis of CO_2 flow patterns essential for heat transfer accenting the features of small diameter channels. Then a method of heat transfer calculation in flow boiling under the conditions typical for carbon dioxide using is suggested. The method is oriented on the regimes forgoing to dryout. In the last section of the paper comparison of the calculation results with available experimental data is given.

FLOW PATTERNS IN CARBON DIOXIDE FLOW BOILING

As is widely known, different researchers distinguish many two-phase flow patterns and rather often they used different names for practically the same flow pattern. Anyone who inspected the photographs of two-phase flow patterns obtained in the experiments has to accept that subjectivism in their interpretation is inevitable in most cases. But from a practical view some distinctions in an actual two-phase structure and all the more in their definitions are not important. In forced flow boiling heat transfer it is the most important to determine such two-phase structures, which correspond to nucleate boiling, to liquid evaporation and to liquid film dryout. In horizontal channels a local wall dryout can be observed at first at the top part of the channel perimeter. At the same time the experiments in small channels, as is reported in [12], most often do not find the stratified flow regimes, especially under conditions of boiling. This has to relate to a greater extent to the conditions of high reduced pressures; as is clear from the papers [2, 6] in the experiments with carbon dioxide boiling the stratified flow regimes have been never observed.

Taitel [11] has suggested to consider bubbly, slug, and churn flow patterns for vertical channel as a joined intermittent flow regime, because the mostly important distinction exists between these structures and annular flow. We used this idea [13] and considered the main two flow patterns under the conditions with the wetted wall of a channel: quasihomogeneous and annular flows. This approach can not be applied obviously to low reduced pressures, but for discussed case of carbon dioxide flow boiling its using seems to be quite reasonable.

The homogeneous model of two-phase flow is justly considered as the simplest and very crude one. But in contrast to merely empirical methods, such as the Martinelli one, it is the model, which becomes rather strict one at the limiting case of a mixture of fine dispersed phases. In this limiting case one can see a certain analogy between the homogeneous model and the model of continuum, which as is known ignores the actual atomic nature of a substance. However, for two-phase flow a problem of determination of a mixture viscosity remains even in the discussed limiting case. At high reduced pressures distinctions in liquid and vapour properties, in densities first, diminish, thus making use of the homogeneous model more defensible. In a widely known book [14] an indicative fact is reported: for water/steam systems at pressures $p \geq 11$MPa (about $0.5p_c$) 95% from 486 experimental points deviate less than 9% from calculations according to the homogeneous model, mass flow rates being in the range 400-3500kg/(m^2 s).

A basic destination of the homogeneous model is calculation of frictional pressure drop. Commonly a wall shear stress is calculated according to the following correlation:

$$\tau_m = (\xi/8)\rho_m u_m^2 = \tau_0(1 + x(\rho_L - \rho_G)/\rho_G). \qquad (1)$$

In this correlation it is assumed that friction factor ξ is computed in accordance with ordinary dependences on Reynolds number, the latter is determined as $\mathrm{Re}_0 = Gd_h/\mu_L$. Homogeneous density ρ_m is calculated using volumetric flow quality β, so $G = \rho_m u_m = \rho_L u_0$, where u_0 is so-called circulation velocity. From the above definitions it is clear that liquid viscosity is used here as a viscosity of two-phase flow. This is reasonable in the quasihomogeneous flow regime when a channel wall is fully wetted. In [13] we have shown that an accuracy of calculations increases if in (1) "actual" mixture density is used. The latter is defined as

$$\rho_\varepsilon = (1-\varepsilon)\rho_L + \varepsilon\rho_G,$$

where ε is void fraction. As usually $\varepsilon < \beta$, so $\rho_\varepsilon > \rho_m$, and as a consequence two-phase friction increases that corresponds an experimentally revealed tendency, at rather high reduced pressures, at least. In horizontal channels there is no physical reason for local slip of the phases, but effective slip arises due to nonuniform distribution of velocity and void fraction in the flow cross-section. As it is shown in [13], for horizontal channels one can assume that $\beta \approx 1.1\varepsilon$ that practically coincides with the well-known Armand formula.

Quasihomogeneous/annular (or intermittent/annular) transition has rather convincing grounding [11, 13, and 14] for upward flow in a vertical channel. But even in this case at high reduced pressures, as is shown in [13], calculations on the method recommended in [11] lead to unrealistic results; in [13] it is suggested to use an additional geometric condition of the discussed transition: $\varepsilon \geq 0.7$. As it was mentioned above, the model of flow patterns transition in horizontal channels by the authors of [10] became obviously invalid for small channels. There are many enough experimental evidences that transition to annular flow regime during boiling in horizontal tubes occurs at higher void fraction than it is predicted by the model. The new flow pattern map for carbon dioxide by the authors of [2] connects the discussed transition with the definite value of vapour quality, which in its turn depends on the ratio of liquid and vapour densities and viscosities. It is surprising that the annular flow establishment does not depend on mass flow rate.

For the purpose of boiling heat transfer prediction for a particular case of CO_2 it is not important to determine exactly the transition to annular flow itself. (It will be argued in the following section of the paper). In this relation more essential transition is replacing nucleate boiling by evaporation of thin liquid film. In these circumstances it is reasonable to use the geometrical condition $\varepsilon \geq 0.7$ as an approximate boundary of an incipience of annular flow in horizontal tubes of small diameter.

HEAT TRANSFER MODEL

At high reduced pressures boiling incipience occurs at very low wall superheats; typical values of an equilibrium vapour bubble radius are very small. For the experimental data used in [6] at the lowest pressure p=1.525MPa (p/p_c=0.21) and ΔT=1.1K radius of the equilibrium vapour bubble R_* $\approx 3.77\cdot10^{-7}$m. At the corresponding to this superheat heat flux $q \approx 1$kW/m^2 temperature variation due to liquid conductivity on such distance is only $2.5\cdot10^{-3}$K, and at the experimentally tested maximal heat flux q = 13kW/m^2 the corresponding temperature variation is $3.25\cdot10^{-2}$K. It is clear that the value of the equilibrium bubble radius is much less than a probable thickness of a liquid film in annular flow regime in the channel of 10mm in diameter where the corresponding experiments have been conducted. The other estimation can be made for the smallest channel from the experimental data cited in [6]. The experimental data on CO_2 flow boiling have been obtained at p $\approx$4.05MPa, q=10 and 20kW/m^2 in aluminum channel of 0.6mm in diameter. The calculated value of minimal vapour bubble radius for the lesser heat flux (corresponding ΔT=1.05K) is approximately $7.2\cdot10^{-8}$m. Bearing in mind annular flow regime with as high void fraction as ε=0.9, one obtains for liquid film thickness $\delta \approx 15\cdot10^{-6}$m. At q=10kW/m^2 the temperature head due to conductivity on this thickness consists ~1.5K, that is 1.4 times higher than in the experiment. It means that evaporation from the film free surface can not transfer the heat flux from the wall at the real wall superheat.

The above estimations show that for carbon dioxide in heated channels the main heat transfer regime is nucleate boiling. This conclusion well consists with many experimental observations [4, 12], which reveal that HTC in CO_2 flow boiling strongly depends on heat flux density and pressure and practically does not on mass flow rate and vapour quality at rather wide range of the vapour quality variation. In many cases the heat transfer coefficient in flow boiling keeps practically the same value at vapour quality from zero up to 0.6 and even 0.8. It is noteworthy to remark that in the last of the above estimations void fraction ε=0.9 corresponds to x=0.75 even in homogeneous approximation. Thus, there is a reason to suppose that due to high reduced pressure nucleate boiling of carbon dioxide occurs even in annular flow regime, liquid film being thick enough for bubble formation. Prof. Thome in [12] advanced his doubts regarding a suggestion that heat transfer in forced flow in small channels is controlled by the mechanism of nucleate boiling. He proposed as a dominant heat transfer mechanism in microchannels a model of elongated bubbles formation and evaporation of thin liquid film on the heated wall under these bubbles. This model appears to be quite reasonable at rather low reduced pressures, but at high pressures it does not. It is remarkable that in the later paper [6] the authors proposed the method of calculation of carbon dioxide heat transfer, which does not use the model of elongated bubbles, the method being recommended both to small and to large channels.

Nucleate boiling contribution into heat transfer to carbon dioxide in [6] is computed according to an empirical formula, which is a modification of the Cooper [15] correlation. The original Cooper correlation is found to be in very bad agreement with the experimental data; its modification has not correct dimensionality and can be used only for particular experimental results on carbon dioxide flow boiling. In [9] the present author proposed an approximate theory of developed nucleate boiling and derived the following predicting equation for heat transfer:

$$q = 3.43\cdot 10^{-4}\frac{\lambda^2 \Delta T^3}{\nu \sigma T_s}\left(1+\frac{h_{LG}\Delta T}{2R_i T_s^2}\right)\left(1+\sqrt{1+800B}+400B\right), \quad (2)$$

where

$$B = \frac{h_{LG}\left(\rho_G \nu\right)^{3/2}}{\sigma\left(\lambda T_s\right)^{1/2}}. \quad (3)$$

The equation is characterized with necessary universality and can be used to carbon dioxide boiling also. In Fig. 1 a comparison of calculated according to (2) and measured in the experiments of [16] HTCs is presented. The experiments have been conducted at rather low (for CO_2) pressures: p=0.738MPa (T_S= -48.1°C, p/p_c=0.1) and p=1.40MPa (T_S= -30.5°C, p/p_c=0.19); horizontal tubes from copper, stainless steel and aluminum with outer diameter of 16mm and different surface treatment were used as heaters. As is seen from the figures, the experimental points are distributed near the computed curves with typical for nucleate boiling scatter, the higher experimental HTCs being observed for surfaces with higher thermal activity $(\rho c\lambda)^{1/2}$ and higher roughness.

As it was noted in the Section 1 of the paper, the available experimental data on carbon dioxide flow boiling have been conducted at rather small heat fluxes. Among the data used in [6] maximal heat flux density is 46kW/m^2 that regards to the experimental run on boiling in the tube of 2mm in diameter at mass flow rate G=1000kg/m^{2}s and pressure p=3.965MPa (p/p_c=0.54). This heat flux consists only 5% from CHF at pool boiling at the same pressure. Such low heat fluxes are rarely met in heat and nuclear engineering, but they are typical for refrigerators and heat pumps. At small heat fluxes a mechanism of single-phase convection can be essential in an entire heat transfer from the heated wall. A simple interpolation is commonly used in order to calculate a total heat transfer coefficient in flow boiling:

$$\alpha = (\alpha_b^3 + \alpha_c^3)^{1/3}, \quad (4)$$

where α_b is calculated on the basis of eq. (2), and α_c is convective HTC. The latter is usually computed according to commonly accepted correlations for single-phase convection. In [13] the Petukhov et al equation is recommended for this purpose.

When one considers an ordinary boiling curve, a range of heat fluxes, which corresponds to commensurable influence of convective and boiling mechanisms of heat transfer, is very narrow. This is due to strong dependence of boiling HTC on heat flux. In many cases, in particular, in applications to power engineering the above range of regime parameters is not important practically. Probably, on this reason effect of convective heat transfer enhancement due to increase of superficial velocity of two-phase flow is not commonly accounted for. An exception is the Chen correlation [17], where an empirical correcting factor is introduced in convective heat transfer contribution in flow boiling heat transfer. As it was mentioned earlier, in the experiments on carbon dioxide flow boiling high values of HTC were

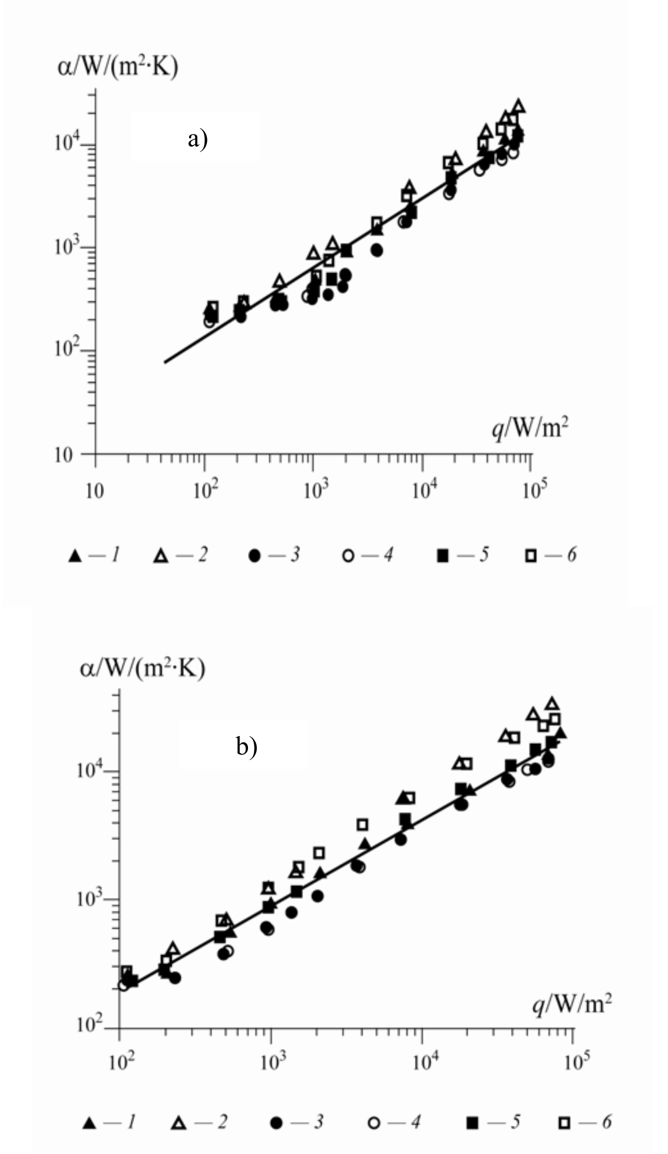

Fig. 1. Calculated and measured HTCs at carbon dioxide pool boiling: lines – calculations according to eq. (2), points – experiments of [16]; a) p = 0.738MPa, b) p = 1.4MPa; 1, 2 – copper grinded and sandblasted tubes, 3, 4 – stainless steel differently grinded tubes, 5, 6 – aluminium grinded and sandblasted tubes.

obtained in the wide range of vapour qualities, i.e. at very different values of the superficial velocity of two-phase flow. At low heat fluxes this leads to increase of convective heat transfer contribution and requires accounting for this effect on total HTC.

It is well known that in two-phase flow the wall shear stress strongly increases with vapour quality increase, eq. (1) approximately reflects this regularity. There are reasons to suppose that intensity of convective heat transfer in two-phase flow also enlarges with greater vapour quality. At turbulent flow regime interdependence between momentum and energy transfer is determined by the Reynolds analogy. This approach has been successfully used for analysis of heat transfer at in-tube vapour condensation [18], the corresponding predicting equation being known as the Boyko-Kruzhilin formula. It can be shown that in turbulent flow for liquids with Pr ≥1 HTC is connected with wall shear stress by simple correlation

$$\alpha \sim \tau^{1/2}. \qquad (5)$$

Using an assumption that at high reduced pressures the homogeneous approximation is justified, one obtains from (1) and (5) for convective heat transfer in turbulent two-phase flow:

$$\alpha_c = \alpha_0 (1 + x(\rho_L - \rho_G)/\rho_G)^{1/2}, \qquad (6)$$

where α_0 is HTC at single-phase convection at the same mass flow rate, which is controlled by $\mathrm{Re}_0 = Gd_h/\mu_L$ and Pr_L. The liquid properties are chosen at the corresponding saturation temperature.

For $\mathrm{Re}_0 \geq 5000$ α_0 is calculated according to the Petukhov et al. equation:

$$\mathrm{Nu} = \frac{\mathrm{Re}_0 \, \mathrm{Pr} \, (\xi/8)}{1 + 900/\mathrm{Re}_0 + 12.7\sqrt{\xi/8}(\mathrm{Pr}^{2/3} - 1)}, \qquad (7)$$

the friction factor is calculated according to the Filonenko formula:

$$\xi = (1.82 \lg \mathrm{Re}_0 - 1.64)^{-2}. \qquad (8)$$

In the range $2000 \leq \mathrm{Re}_0 \leq 5000$ the modification of eq. (7) made by Gnielinski is used.

Thus, convective HTC calculated in accordance with equations (6-8) is used in eq. (4) in order to obtain the total HTC in flow boiling of carbon dioxide. Such calculation method allows to reflect the actual small increase of HTC with vapour quality observed in the experiments at low heat fluxes and rather high mass flow rates.

In many experimental studies of carbon dioxide flow boiling heat transfer the regimes with dry wall have been obtained. As it was noted in the Section 1 of the present paper, the correlation of [2] for determining dryout incipience can not be accept from physical view, it allows only to describe some particular experimental data. Basing mostly on the data for water/steam systems Sergeev [19] has developed a general correlation for predicting so-called boundary vapour quality:

$$x_b = 1 - 0.86\exp(-19/\mathrm{We}_L^{1/2}), \qquad (9)$$

where $\mathrm{We}_L = G^2 d_h/(\rho_L \sigma)$. This correlation was grounded by means of qualitative analysis of main mechanisms in thin liquid film at the annular regime of two-phase flow and justified by wide comparison with experimental data not only for the water/steam system. We have tested the correlation on the data for carbon dioxide and found that it failed to incorporate the actual regularities of the process. This seems to be natural as dryout in the experiments occurred at unusually low heat fluxes, but eq. (9) does not account for the heat flux effect. Qualitative analysis of the data shows that this effect does exist. Besides, some experiments are featured with rather gradual but not abrupt decrease of HTC with vapour quality, the latter being in contrast to the Sergeev's model of dryout. The issue on actual mechanism of dryout in the considered conditions remains the open one. In comparison with the experimental results we confined ourselves by the data regarded to the regimes with the wetted wall.

However, an attempt was made to account for a possible local dryout at the channel wall. Annular-dispersed flow is considered; a possibility of the local dryout is connected with two circumstances. The first one is evaporation of liquid droplets deposited on the wall surface; the second one is nonuniformity of a liquid film thickness due to gravity, this refers only to the horizontal channels. Dryout is assumed not to occur if liquid mass flow rate due to the droplets deposition (m_d) exceeds mass flow rate due to evaporation. Thus, an equality $m_d h_{LG} = q$ corresponds to the condition of dryout incipience, and a fraction of dry surface can be estimated as

$$Y \sim q/(m_d h_{LG}). \qquad (10)$$

The liquid drops deposition is considered as a result of turbulent diffusion, which can be estimated as the effective Reynolds mass flux multiplied by the droplets mass fraction in the vapour flow. The latter is assumed to be proportional to the liquid mass fraction $(1 - x)$ in the channel cross-section of the interest. Consequently, the liquid mass flow rate due to the droplets deposition can be expressed as follows

$$m_d \sim G(1 - x)(\xi/8)/(1 - 12.7(\xi/8)^{1/2}),$$

and the fraction of dry surface as follows:

$$Y = kq(1 - 12.7(\xi/8)^{1/2})/(G h_{LG}(1 - x)(\xi/8)). \qquad (11)$$

Numerical factor k, characterizing amount of ignorance of the real mechanism of the process, is assumed to be equal to unit in our calculation. (This can be considered as an argument in favour of validity of the approach).

Heat transfer at the dry surface is assumed to be controlled by single-phase convection to the saturated vapour. A characteristic length l_d of the circle dry spot can be estimated according to the following simple correlation $\pi d_h l_d Y \approx \pi l_d^2/4$. So that $l_d \approx 4Yd_h$. In single-phase convection in laminar boundary layer at constant wall heat flux a nondimensional local HTC is calculated as follows

$$Nu_l = 0.47\,\mathrm{Re}_l^{1/2}\,\mathrm{Pr}_v^{1/3}, \qquad (12)$$

where l is a distance from the front edge of the surface. As $\alpha_v \sim l^{-1/2}$, the HTC averaged on the full length of the dry spot l_d is doubled value of the HTC at the end edge of the spot. Bearing in mind the above estimation of l_d one can use eq. (12) for calculating the average HTC from the dry spot to vapour, if $l = Yd_h$. In eq. (12) the saturated vapour properties are used, the Reynolds number $\mathrm{Re}_l = Gxd_h Y/(\mu_v \varepsilon)$, the Nusselt number $Nu_l = \alpha_v d_h Y/\lambda_v$. The vapour/ liquid slip is accounted for with accordance to the method described in the section 2.

On the wetted surface HTC is computed according to eq (4), the eqs (2) and (3) being used for boiling HTC α_b and eqs (6-8) being used for convective HTC α_c. The wall superheat at the dry surface is determined as $\Delta T_{dry} = q/\alpha_v$, and at the wetted surface the wall superheat is calculated as $\Delta T_{wet} = q/\alpha$. The average wall superheat is determined accounting for the corresponding fractions of the surface, so that an effective HTC is calculated as follows:

$$\alpha_{eff} = q/(Y\Delta T_{dry} + (1 - Y)\Delta T_{wet}). \qquad (13)$$

This approximate and simple method to account for an effect of the partial wall dryout seems to be in reasonable agreement with experimentally observed tendencies. As $\alpha_v \sim (Yd_h)^{-1/2}$, at low flow qualities, when the dry spots are very small, the partial wall dryout does not affect the effective

HTC. A hydraulic diameter increase leads to lower α_v and to decrease of α_{eff}. At this stage comparison of calculated values of α_{eff} with the experimental data has been conducted at k=1 in eq (11) and without any additional corrections; the results will be presented in the following section of the paper.

COMPARISON WITH THE EXPERIMENTAL DATA

Due to a kindness of Prof. J. Thome and Dr. Cheng L. from Switzerland we could use all the experimental data on carbon dioxide flow boiling, which have been analyzed in their paper [6]. These data have been obtained during the last decade, mostly after 2000 and have been published in 16 papers cited in [6]. The majority of experiments have been conducted in the channels of small hydraulic diameter, $d_h \leq 3$mm, three groups of the data related to the "ordinary" channels (6 – 10mm in diameter). In all cases the channels were the horizontal ones. The reduced pressures range was p/p_c= 0.21 – 0.87, the saturation temperature were (-28) - 25ºC.

We tested the described above method of HTC calculation on all available data. A general conclusion is that the proposed method well agrees with all the data related to the regimes preceding to dryout; according to [6] this consists 773 points (from 1124 points at all). Several examples of the comparison of the calculated (lines on the figures) and the measured (points) HTCs are presented below.

In the figures 2a and 3 the data of [20] and [21] at low heat fluxes are presented. As is seen, essential dependence of HTC on the vapour quality, especially at the lower pressure and the higher mass flow rate (Fig.2a) is observed. For this case $\alpha_b \approx 7.4$kW/m^{2}K, this is close to the experimental values only at the low vapour qualities ($x \leq 0.15$); at higher x HTC increases more than two times. Eqs (6) and (4) successfully reflect this effect. At the higher pressure (T_S=15°C) increase of HTC with x due to convective heat transfer is not so significant, because at this pressure and q=9kW/m^2 boiling HTC is as high as $\alpha_b \approx 13$kW/m^{2}K. Fig. 2b presents the data of [20] at the same pressure and the channel as in Fig. 2a, but at essentially higher heat flux (q=20kW/m^2). As a result, influence of convective contribution in heat transfer is less even at high mass flow rate G = 1500kg/m^{2}s. Under the conditions of the experimental runs shown in the figures 2 and 3 effect of the partial wall dryout is very small, practically negligible, the calculated lines for α_{eff} and for α coincide. Good agreement of the prediction and the experimental data is observed, practically all points being in the band ±30% denoted with the dotted lines equidistant from the predicted line.

The vertical dotted lines in the figures correspond to a boundary vapour quality x_b calculated in accordance with eq (9). In the case of Fig. 3 (p/p_c=0.69) one can say that the experimental results confirm prediction, as the author of [19] has determined an accuracy of the equation as ±20%. According to (9) decrease of x_b with growth of mass flow rate has to be observed. But at the lower pressure (p/p_c=0.54) and at the higher mass flow rates, as it is seen from Fig. 2, wall dryout does not occur even at the vapour qualities much higher than the predicted ones.

A noticeable effect of the partial wall dryout is predicted according to the developed method only for the channels of ordinary size (6-10mm). For example, in Fig. 4 the "eldest" experimental data on CO_2 flow boiling [22] are presented (p=1.525MPa, d_h=10.06mm, G=80kg/m^{2}s). The predicted values of HTC decrease with growth x up to 0.8 approximately 10% at q=8kW/m^2 and 20% at q=13kW/m^2 in spite of some increase of convective constituent of heat transfer. As is seen from Fig.4, this is in excellent agreement

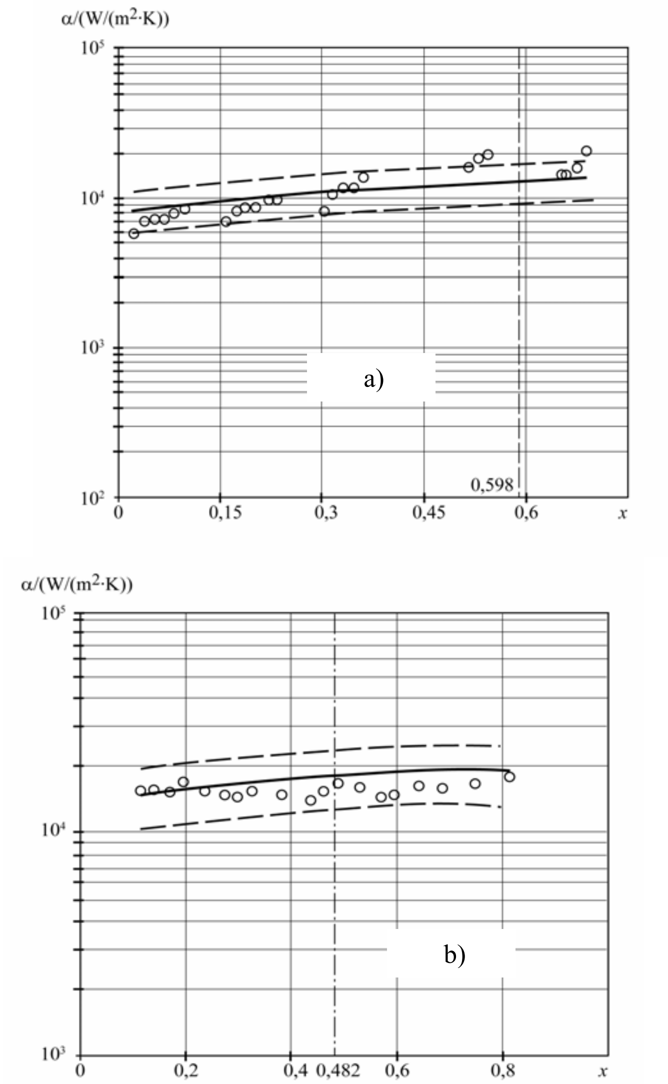

Fig.2. Calculated and measured in [20] HTCs at carbon dioxide flow boiling: p=3965kPa, d_h=2mm; a) q=7.2kW/m^2, G=1000kg/m^{2}s, b). q=20kW/m^2, G=1500kg/m^{2}s.

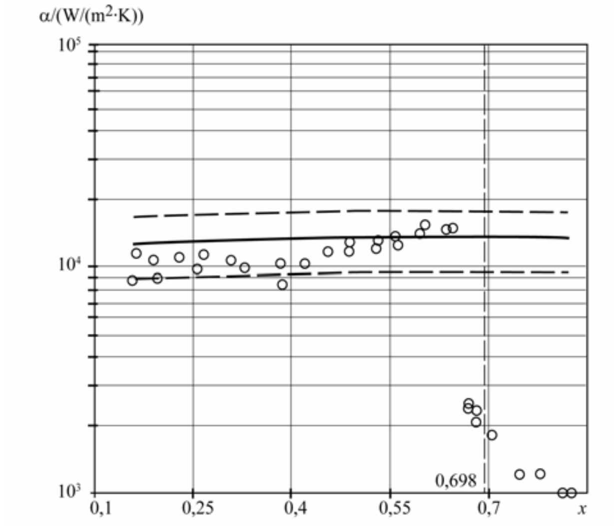

Fig.3. Calculated and measured in [21] HTCs at carbon dioxide flow boiling: p=5069kPa, d_h=1mm, G=720kg/m^{2}s, q=9kW/m^2.

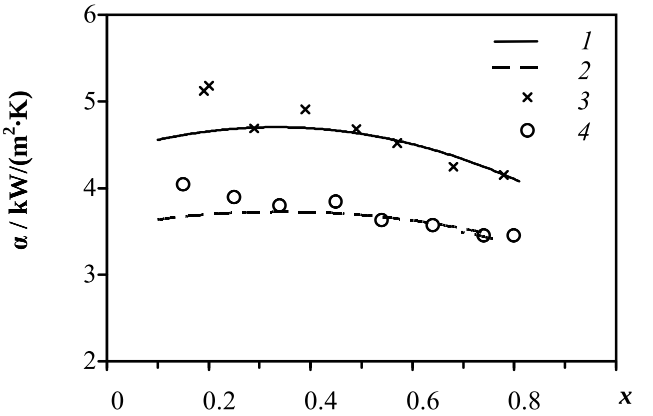

Fig.4. Calculated and measured values of HTC at carbon dioxide flow boiling: p=1525kPa, d_h=10.06mm, G=80kg/m^{2}s, 1, 3 - q=13kW/m^2, 2, 4 - q=8kW/m^2 [22].

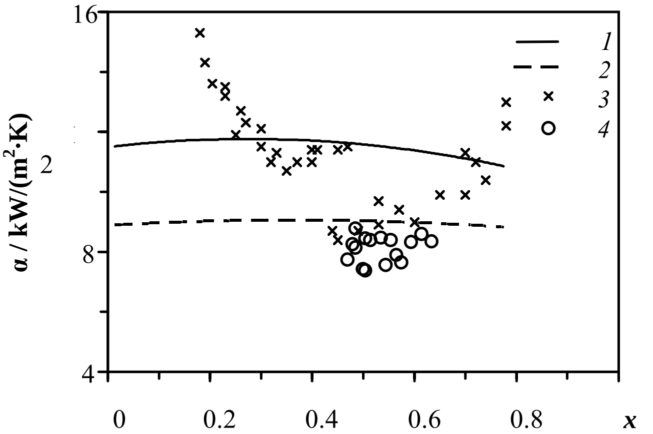

Fig.5. Calculated and measured values of HTC at carbon dioxide flow boiling: p=2881kPa, d_h=3mm, G=590kg/m^{2}s, 1, 3 - q=21kW/m^2, 2, 4 - q=10kW/m^2 [25].

with the measurements of [22]. The similar results are found in comparison of the predicted HTC with the measured in [23] and [24], where circular tubes 6 and 7.53mm were used as heated channels. In these studies essentially higher pressures were tested (p=3.48-5.7MPa).

For intermediate channel size and pressure (d_h=3mm, p=2.88MPa) comparison with the data [25] is given in Fig. 5. Both predicted curves and the data do not demonstrate noticeable influence both of the partial wall dryout and of convective heat transfer increase with flow quality increase. As for the data of [26] in flow boiling in the smallest channel (d_h=0.6mm) at p=4.03MPa, G=400kg/m^{2}s and q= 10 and 20kW/m^2, in spite of the scatter of the points, satisfactory agreement of the prediction and the was observed. The experimental points tend to lie lower the predicted lines only at x>0.8.

It is to be noted that there are some contradictions in the available experimental data. In particular, in the experiments of [27] at relatively high heat flux q=32kW/m^2 and low mass flow rates (G≤260kg/m^{2}s) in the tube 1.8mm in diameter at 3 different pressures heat transfer intensity did not only fall, but even grew up to x=0.82, while at similar conditions, but at the higher mass flow rates in [20] decrease of HTC values was observed beginning from $x \approx 0.4$. It is difficult to explain decrease of HTC values with pressure increase at p>4.5MPa at the same other conditions in the paper [24]. Due to very small wall superheats in carbon dioxide flow boiling it is extremely hard problem to guarantee high accuracy in the measurements, especially in multi-channel devices. Probably, one must not overestimate significance of every experimental point under these conditions.

The above analysis has shown that the developed method based on universal enough approaches is suitable for predicting heat transfer in carbon dioxide flow boiling. In principle, there is no difficulty to achieve the better results in comparison with the available experimental data by means of correction of numerical factors in eqs (11) or (12). However, it seems to be more reasonable to test the developed predicting method on other substances at the similar reduced pressures. This will be undertaken in the nearest future.

CONCLUDING REMARKS

1. Analysis of available experimental data on carbon dioxide flow (and pool) boiling heat transfer has revealed that their discrepancy with existing correlations is not a consequence of any particular properties of the substance, but a result of a lack of physical basis of the correlations. In reality, high reduced pressures, which are a feature of CO_2 flow boiling data, do not impede, but rather promote in heat transfer modelling. The level of reduced pressures is a reason of high HTC values in CO_2 boiling in comparison to other refrigerants boiling HTCs.

2. Due to high reduced pressures the main heat transfer regime for carbon dioxide in heated channels is nucleate boiling, which dominates at low heat fluxes even at annular flow pattern. The eq (2) proposed by the author is shown to be in good agreement with the experimental data on carbon dioxide boiling.

3. Contribution of convective mechanism into total heat transfer can be remarkable at low heat fluxes, when nucleate boiling keeps up to high vapour qualities. Under these conditions a role of convective heat transfer increases with vapour quality increase; this effect has been involved in the predicting equation for the first time in the present paper.

4. An approximate method has been developed to account for the local wall dryout at carbon dioxide flow boiling; the method satisfactorily describes the experimental tendencies of heat transfer variation in the channels of 6-10mm in diameter.

NOMENCLATURE

B nondimensional number according to eq. (3)
c specific heat, J/(kg K)
d diameter, m
G mass flow rate, kg/(m^2 s)
h_{LG} latent heat of evaporation, J/kg
k numerical factor, nondimensional
l characteristic linear scale, m
m_d effective Reynolds mass flux, kg/(m^2 s)
Nu Nusselt number, nondimensional
p pressure, Pa
Pr Prandtl number, nondimensional
q heat flux density, W/m^2
R_* equilibrium vapour bubble radius, m
Re Reynolds number, nondimensional
R_i gas constant (individual), J/(kg K)
T temperature, K
U velocity, m/s
We Weber number, nondimensional
x flow quality, nondimensional
Y fraction of dry surface, nondimensional

Greek symbols
α heat transfer coefficient, W/(m^2 K)
β volumetric flow quality, nondimensional
$\Delta T = T_W - T_S$ wall superheat, K
δ liquid film thickness, m
ε void fraction, nondimensional
λ thermal conductivity, W/(m K)
μ dynamic viscosity, kg/(m s)
ν kinematic viscosity, m^2/s
ξ friction factor, nondimensional
ρ density, kg/m^3
σ surface tension, J/m^2
τ shear stress, Pa

Subscripts
b boiling, boundary
c critical, convective
d droplet, dry
eff effective
G gas (vapour)
h hydraulic
L liquid
m mixture
S saturation
tp triple point
v vapour
w wall
ε actual
0 refer to single-phase liquid flow

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