MODELLING OF TWO-PHASE FLOW PATTERNS IN A BED OF SPHERICAL FUEL MICROELEMENTS

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Abstract

The paper considers the problems of flow boiling in a bed of spherical fuel microelements. Existing in this field experimental results relate mainly either to pool boiling at low reduced pressures or to two-phase flow in porous media with submillimetric channels at very low mass velocities of coolant. Flow boiling in microchannels, which became an object of intensive investigations during the last decade, is analyzed as a phenomenon similar to flow boiling in tortuous channels of porous space of the bed. It is shown that using any empirical or semiempirical correlations based on the experimental information obtained with conventional channels has no perspective. The significant linear scales as vapour bubble departure in flow boiling and liquid drop diameter in annular-dispersed flow are estimated in comparison with a channel cross size. A model capillary is proved to be a useful tool for modelling of flow boiling in a bed of internally heated particles at high reduced pressures. Under these conditions an intermittent and annular flow patterns are the most significant for two-phase flow regularities. Some new approaches to analysis of these flow patterns are proposed in the paper. As an example, pressure drop, void fraction distribution and some internal characteristics of steam-water two-phase flow in a bed of internally heated metallic spheres at 7.0 MPa are calculated. The calculations show a possibility of stable cooling of such bed even at very high level of power generated.

KEYWORDS

Minichannel, microchannel, single-phase flow, two-phase flow, stabilized heat transfer, boiling incipience, bubble departure diameter, flow boiling, two-phase flow pattern, fuel microelement, modeling capillary, intermittent flow pattern, annular flow.

INTRODUCTION

Recently an idea of water-cooled nuclear reactor with spherical fuel elements becomes a subject of scientific discussion [1, 2]. Till now only gas-cooled reactors with spherical fuel elements have been experienced [3]. One can accept a statement of [1] that in a case of single-phase water cooling of spherical fuel elements no apparent scientific and technical problems appear. But in a case of boiling water type reactor with spherical fuel microelements a number of issues arise in relation to hydrodynamics and heat transfer regimes. Available information on boiling in a bed of the spherical heat-generating elements is limited by several works. Some of them was initiated by the accident at the Three Mile Island nuclear power plant and were published in 1980th (for example, [4, 5]). The main problem studied was a possibility of passive cooling of radioactive debris. Induction heating of metallic spheres about 3 mm in diameter was used in these studies, the experiments were performed at near atmospheric pressure. The investigations in this direction are continued, as it is shown in the recent paper [6]. The latter presents the experimental data on cooling of a porous bed with internal heat sources by means of refrigerant R134a boiling within the pressure range corresponding to maximum value of pool boiling CHF. Rather high pressure and the liquid supply from the bottom of the bed provided its good coolability. But all above studies dealt with pool boiling or pool boiling with an additional possibility of natural circulation [6]. Although the later experimental work [7] is devoted to the other application, it also examined pool boiling in fuel particles at atmospheric pressure.

For the prospective water cooled nuclear reactor forced flow boiling in a bed of fuel elements is of interest. The paper [8] presents important experimental results and comprehensive enough review of this problem. Flow boiling in porous structures with internal heat generation was studied in connection with the problems of transpiration cooling and heat transfer enhancement. In the both cases

submillimetric porous particles and rather low mass velocities of coolant were the subject of investigations. Nevertheless, some experimental results and their physical interpretation seem to be important for the problem discussed. In particular, the authors of [8] reasonably believe that homogeneous two-phase flow structure in interstitial space is of low probability and the model of separated flow is more suitable at low and moderate pressures. However, the Lokhart-Martinelly predicting correlations can hardly be considered as physically grounded approach here. They are only a method of experimental data treatment, not more. The experimental results of [8] on boiling heat transfer at the surface of the heated wire placed in the porous structure and, in particular, dependence of heat transfer coefficient on two-phase flow quality can be useful for modelling of flow boiling in the porous bodies with internal heat sources. Certainly, these results are hardly applied directly for this purpose.

Evidently, experimental and theoretical studies of hydrodynamics and heat transfer of two-phase flows in the channels of a small hydraulic diameter can give important results for better understanding main mechanisms of momentum and energy transfer in two-phase flows in a bed of spherical fuel elements. A direct capillary presents the simplest, but often used model for simulation of hydrodynamics in tortuous channels in a porous space. Certainly, actual phenomena in the porous space are essentially more complex due to continuous variation of flow direction and of the channel cross section. Besides, the channels in capillary-porous bodies are interconnected, and a transverse permeability of isotropic porous bodies does not differ from the permeability in flow direction, this creates an additional complexity in the analysis. Nevertheless, the simulation of two-phase flow in porous space by means of a single channel flow can give some useful and more detailed information on the process than using the integral predicting methods based on the Darcy approximation. As for the interchannel connections, the experimental investigations of two-phase flow in the parallel microchannels can shed light upon the problems of the interest.

Flow boiling in minichannels and microchannels became the subject of very intensive examination rather recently, practically during the last decade. Although the initial stage of these investigations when some contradictious and even doubtful results have been published can be considered as overcome one, many issues of two-phase flow patterns, pressure drop, heat transfer and boiling crisis in mini- microchannels have not been understood till now.

In this paper first of all a short analysis of state of the art in two-phase flow and heat transfer in mini- microchannels is presented. To the opinion of the present paper authors, the existing classifications of small diameter channels as mini- or micro- are not physically grounded. It is clear that any absolute value can not be used as a boundary in such classifications. There is a need here in a scale of such kind as the Laplace constant in gas-liquid systems in the hydrostatics problems or a molecule free path in the dynamics of rarefied gases. In any case the same value, for example, 1 mm is strongly different scale for water-steam mixture at the pressures 1 bar and 100 bar. In the analysis a particular attention is payed to those experimental results, which relate to rather high reduced pressures when a ratio of a liquid density to its vapour density is not very high.

High reduced pressures force to change an ordinary view on two-phase flow regularities in mini-microchannels. Really, for water at pressures 5-10 MPa equilibrium ("critical") size of a steam bubble is of the order of tenths of micrometer, the Laplace constant is about 1.4-1.7 mm. This gives some reasons to consider capillaries 1-2 mm in diameter as ordinary tubes for water boiling at high pressures. Basing on this assumption we performed an analysis of steam-water two-phase flow in a bed of spherical heat-generating elements. Actual flow through the bed is simulated by flow in a direct circular tube with diameter equal to the hydraulic one of the porous space. Some modification is proposed in the calculations of void fraction and pressure drop in two main flow patterns considered in the framework of the model (intermittent and annular). An example of the analyses has been presented in the paper for a bed of internally heated particles with rather high value of heat flux ($1MW/m^2$) at the metallic sphere surface. Calculations showed a possibility of stable cooling of such particle bed by means of flow boiling water at pressure 7 MPa. Pressure drop in the bed at calculated regimes appeared to be not very large.

THE MAIN REGULARITIES OF FLOW AND HEAT TRANSFER IN MINI-MICROCHANNELS

The recent comprehensive paper [9] on flow boiling in channels of small diameter became now very popular and is often referred in modern publications. In this paper it is proposed to establish the

boundaries between small diameter channels, minichannels and microchannels basing "on engineering practice and application areas employing these channels". According to [9] the following ranges of hydraulic diameters (d_h) are attributed to different channels: conventional channels with $d_h \geq 3$ mm, minichannels with d_h =0.2-3 mm, and microchannels with d_h=0.01-0.2 mm. This approach can be accepted only from view of some practical convenience. In particular, exacting requirements to high purity of liquid are very important namely for the microchannels (according to the Kandlikar [9] classification). But in relation to intrinsic features of two-phase flow it is rather doubtful that hydraulic diameter d_h =0.2 mm corresponds to any essential variations in mechanisms of momentum and energy transfer. In this view a definition of microchannels as limited by d_h=0.01 and 1.0 mm used in [10, 11] has not less (but not greater) basis than the Kandlikar classification.

It seems to be very important not to be deceived by false criteria and to escape some fetishism in this problem connected with conceptions of mini- and microchannels. Really, a cross size of a channel is a significant parameter, which strongly affects flow regime, pressure drop and heat transfer in single-phase flow. In particular, at stabilized laminar heat transfer in round tubes heat transfer coefficient (HTC) is inversely proportional to a tube diameter. Extremely high HTCs obtained in the experiments with microchannels are, consequently, quite natural. But there are no grounds to anticipate any qualitative changes in flow and heat transfer with a channel diameter decrease, while a model of continuum remains valid. It means practically that the general regularities of single-phase flow have to be valid at least at Knudsen numbers Kn<0.01. For practical application this condition is always fulfilled for liquids and for gases at atmospheric and higher pressures.

Obviously, in single-phase flow and heat transfer in small diameter channels there are no scientific problems. But there are many engineering difficulties in experimental studies connected with a channel wall roughness, coolant purity, methods of measurements and their result interpretation. If these difficulties are successfully overcome, the experimental data on pressure drop and heat transfer at laminar flow in channels with d_h<1 mm fully agree with conventional analytical predicting formulas [12-14].

In contrast to single-phase flow a channel size influence on two-phase flow hydraulics and heat transfer can be more strong and complicated. The presence of interfaces not only requires to take into account in analysis a surface tension force, but leads to appearance of additional internal linear scales, which can be commensurable with a channel size. This means that consideration of flow boiling in small channels actually demands to juxtapose the channel diameter with such internal scales of two-phase flow as a vapour nucleus size, a vapour bubble departure diameter, a liquid drop diameter, and, probably, a wave length at the interface. In this way it is possible to obtain some physically grounded criteria for the classification of channels on their size. To the best knowledge of the present authors there were no attempts till now to obtain such criteria. In the paper [9] some comparative estimations of characteristic sizes of a vapour bubble and a channel are given, but they are not used for the classification of the small channels.

Practically three main issues are significant for an analysis of flow boiling in mini- microchannels:

- nucleate boiling incipience in the small channels;
- two-phase flow patterns and the conditions of transition from one pattern to another;
- boiling crisis conditions.

As for HTC in flow boiling in the small channels, its value is high enough even at laminar flow of a single-phase liquid. Really, in a channel of d_h=0.2 mm the stabilized value of HTC for water at laminar flow regime and constant heat flux at the wall is about 14 kW/(m^{2}K). Thus, before boiling crisis any problems with heat transfer intensity in microchannels hardly arise.

A detailed analysis of boiling incipience in small channels flow is given in [9]. First of all it is essential to note that nucleate boiling of water has been detected experimentally by video camera even in the channel of 40 μm in diameter. This is quite natural because a vapour nucleus size for water at atmospheric pressure is about few micrometers and becomes much less at the higher pressures. On the other hand, high values of HTC in single-phase liquid flow in microchannels mean that rather high heat fluxes can be removed from the heated wall without boiling. The paper [15] gives a prognosis on rather high wall superheats at boiling incipience in the channels smaller than 50 μm in diameter. At the same time it is clear that in microchannels a saturated liquid has to be superheated at the rather small length (together with the wall). The conditions similar to homogeneous nucleation have to be realized in this case. The experiments of [12] demonstrated such a possibility: at low heat fluxes (less than 3 kW/m2) up to 70 K liquid superheat over the saturation temperature was observed in convective

boiling of HCFC123 and FC72 in 0.19 and 0.51 mm ID tubes. On the other hand, normal saturated boiling was observed with a higher heat flux of q=12.6 kW/m^2 in the channel of 0.51 mm in diameter and at q=5.5 kW/m^2 in the 0.19 mm ID tube. The above high liquid superheat is probably explained by the conditions of high liquid purity and very small roughness of the channel wall, which are necessary for the experiments with microchannels. There is no explanation of the reason of easy boiling incipience at the higher heat fluxes, i.e. at higher a liquid temperature gradient in the axial direction. However, in view of practice the latter results are more natural, since it is very difficult in an experiment to achieve the conditions of homogeneous nucleation. Thus, one can believe that boiling incipience in microchannels does not require too large wall superheat.

After nucleate boiling onset a bubbly flow regime (regime of isolated bubbles) can be observed in microchannels [9]. But in small channels a vapour bubble is able to cover an entire cross-section of the channel before departure. This evidently leads to the regime of confined vapour bubbles [9] or to slug regime of two-phase flow. A bubble departure diameter is, consequently, a significant parameter determining two-phase flow pattern in a microchannel.

As it is shown in [16, 17], the strict analytical solution of the problem of a bubble departure in nucleate boiling is impossible. The published numerical results in this field were obtained for some particular conditions at pool boiling. At the same time in [17] an approximate approach to the problem of vapour bubble departure in flow boiling has been grounded and successfully applied to turbulent flow. According to this approach a vapour bubble detaches from its nucleation site and begins to slide along the channel internal surface when the bubble growth rate decreases to the local flow velocity at the distance $y=R$ from the wall. At moderate and high reduced pressures a bubble growth rate is governed by heat supply to the interface. This thermal diffusion approximation is practically valid at Jakob number Ja≤200. In this case according to [17] a vapour bubble growth law is as follows

$$R = (0.3Ja + \sqrt{0.09Ja^2 + 12Ja})\sqrt{\kappa t}. \tag{1}$$

For convenience we will designate the expression in the brackets in eq. (1) as nondimensional growth module A, so that this equation transforms to very simple form: $R = A\sqrt{\kappa t}$.

For turbulent flow in [17] the equation for bubble departure radius is given first in a general form as follows

$$\mathrm{D_d/d_h = k(A^2/(Re^2\,Pr\,\xi_0))^{1/3}}\ , \tag{2}$$

with numerical factor k≈1.1. For practical use it is more convenient to have also the specific equations for high and rather low reduced pressures, in which asymptotic dependencies $q(\Delta T)$ were used. These equations according to [17] are

$$D_d/d_h = 4.25(q\nu/(h_{LG}\sigma))^{1/9}/(\mathrm{Re}^2\,B\xi_0)^{1/3}, \tag{3}$$

for high reduced pressures (p/p_{cr}>0.01);

$$D_d/d_h = 4.0(q\nu/(h_{LG}\sigma))^{2/9}(\mathrm{Pr}/(\mathrm{Re}^2\,B^{4/3}\xi_0))^{1/3}, \tag{4}$$

for low reduced pressures (p/p_{cr}<0.01). Nondimensional parameter B reflects individual properties of a fluid and strongly depends on absolute pressure. Reynolds number is calculated on the basis of liquid single-phase mass flow, viscosity is determined at saturation temperature; d_h is used as a linear scale. The friction factor is calculated according to the Filonenko formula:

$$\xi_0 = (1.82\lg\mathrm{Re} - 1.64)^{-2}. \tag{5}$$

The above equations allow determining a condition of transition to the regime of slug flow in small channel. If bubble departure radius becomes equal or greater than a characteristic cross size of a

channel, this means that boiling onset immediately leads to slug flow regime. In this case a bubbly flow pattern can be observed only at very short interval of the channel length or can not be observed at all. Thus, assuming in the equations (2-4) $D_d=d_h$ one can obtain an approximate condition of the transition discussed.

It is possible to apply the approach of [17] to laminar main flow, which was the most frequent case in the available experiments in microchannels. Bearing in mind proximity of the estimations let us assume a linear velocity profile near the wall, but the wall shear stress is determined by the Poiseuille law. This gives for around tube

$$u = 4u_0 y / r_0 . \tag{6}$$

Differentiating eq. (1) and equating the obtained bubble growth rate to flow velocity in accordance with (6) at $y=R$ one can find for bubble departure diameter

$$R_d = A\sqrt{\frac{\kappa r_0}{8u_0}} . \tag{7}$$

Assuming that the condition of to slug flow pattern transition is $R_d=r_0$ one obtain the following criterion of this transition

$$\frac{A}{2\sqrt{Pe}} \geq 1 , \tag{8}$$

where Peclet number $Pe=u_0 d_h/\kappa$. As far as possible to estimate on the basis of the experimental results presented in [13] and [18, 19] in these studies the inequality (8) was not satisfied near boiling incipience, and a bubbly flow pattern of two-phase flow did exist. The transition to slug flow pattern was probably connected with the bubble coalescence and occurred at small but finite flow qualities. As it is clear from the equations (2-4), at turbulent flow there is a greater possibility to have a bubbly flow pattern in microchannels. At the same time at low pressures (with high Jakob numbers) a slug flow pattern can be appeared in the cross section of boiling onset even in the tubes of conventional size. Thus, communicated in [9] existence in mini- and microchannels not only slug and annular, but isolated bubbles flow regime can be explained with the aid the above approximate analysis of bubble departure conditions.

The boundaries between flow patterns in two-phase flow and conditions of transition from one pattern to the other are determined on the basis of empirical or semiempirical correlations. The mostly physically grounded model of flow pattern transition by Taitel [20] is based on rather simple and clear enough considerations, but this model is inevitably (as any model for two-phase system) approximate one and includes some experimental information. This is natural that some points of the Taitel model can not be applied to the conditions of microchannels. In particular, according to [20] the transition to annular flow in horizontal channel is controlled by the mechanism similar to the Helmholtz instability development. An inertia force of vapour flow is considered as a disturbing one, and a gravitational force is the only restoring force. Such approach is probably justified for rather large channels, but in channels with diameter much less than Laplace constant b a surface tension force is much greater than the gravitational one. Using the Taitel model of annular pattern for two-phase flow in microchannels in the papers [10, 18, 19] seems to be very doubtful, in spite of good results of the model application to the experiments of the authors of these papers.

Probably, the Taitel model of transition to annular pattern in vertical channel is also hardly applied to two-phase flow in microchannels. This transition is connected with achievement of such value of vapour superficial velocity, which is twice terminal velocity of falling liquid drops. However, it is difficult to say on liquid drops in microchannels in an ordinary sense. If one applies a commonly used estimation for a drop size basing on critical Weber number, in the experiments of [10, 18, 19] or [13, 15] diameter of water drops in annular flow can be as high as 1 mm or more. Under these conditions using some empirical function for drop deposition in the model of [18, 19] seems to be also doubtful.

Here one meets a typical case of using some semiempirical approach out the limits of its validity. The matter is not the unusual features of two-phase flow in microchannel, but an absence of strict mechanistic models for two-phase flow and heat transfer. The latter point relates to conventional channels also. The comparison of the different predicting correlations for pressure drop and heat transfer in two-phase flow with the experimental data on microchannels presented in the papers [10, 18, 19] and the very recent ones [21, 22] appears to have a little sense. This is even more so because some of the above correlations relate to turbulent flow, while the experimental data have been obtained at Reynolds numbers less than 300. Thus, to our view, the experimental studies of flow boiling in microchannels did not reveal some unexpected regularities, but shed light on a lot of not properly understood problems in two-phase thermal hydraulics.

As for boiling crisis in small channels, few experimental data were obtained specially in this field. At the same time extremely high critical heat fluxes (CHF) have been obtained in boiling of subcooled water in the channels of small diameter as early as in the beginning of the 1960th. Recently [23, 24] systematic data on ultrahigh CHF at water boiling in channels of 0.406 and 0.902 mm in diameter were presented. High subcooling and high flow velocity allowed obtaining CHF up to the record value of 276 MW/m^2 at pressure 3.1 MPa. As it was shown in [25], under moderate and high reduced pressures there is no difference in boiling crisis mechanisms in tubes of submillimetric and conventional diameter. It is pertinent to note that in [21, 22] in saturated refrigerant R134a boiling at low flow velocities heat fluxes up to 938 kW/m^2 have been obtained under pressure about 0.6 MPa in the microchannel. However, one need to bear in mind that at low pressures extremely high vapour velocities can be achieved in microchannels. This is rather surprising, but the authors of [13] did not pay attention that practically in all their tests boiling crises occurred at vapour superficial velocity close to the steam sound speed.

In Introduction it was mentioned that at high reduced pressures the difference between two-phase flow in micro- and conventional channels has to vanish. Several papers (for example, [26-28]) are devoted to refrigerants two-phase flow in small channels at moderate and high reduced pressures (up to 0.49). The fact that existing correlations are not valid to predict transition of two-phase flow pattern [26] and pressure drop [27, 28] is natural enough due to absence of strict mechanistic models in this field. At the same time it seems to be important that in [26] all usual two-phase flow patterns were visually observed in vertical round tubes of 2.01 and 4.26 mm in diameter for refrigerant R134a under pressures 6-14 bar and in the wide range of superficial velocities of vapour and liquid. The authors of [26] justly pointed out significance of pressure for flow pattern transitions. However, it is difficult to agree with them in relation of possibility to use absolute pressure as a controlling parameter in predicting correlations. It is well known that conservation equations contain only derivatives of pressure and influence of absolute pressure has to be appeared through variation of liquid and vapour properties.

It is well known that the similarity theory requires to derive dimensionless parameters (similarity numbers and criteria) grounding on the mathematical description, i.e. the conservation equations and initial and boundary conditions. In two-phase systems due to the interface presence derivation of a closed mathematical description meets the insuperable difficulties. In these conditions a common way of deriving of nondimensional parameters is based on qualitative analysis and the Buckingham π-theorem using. This leads inevitably to arbitrary decisions. The scientists of the elder generations remember that in the sixtieth of the last century a number of nondimensional parameters for nucleate boiling modelling were commensurable with a number of investigators in this field. This immemorial problem has not been overcome till now. In particular, in [15] two new nondimensional parameters are proposed for flow boiling in microchannels basing on rather doubtful approach.

To our view there is no ground to invent new nondimensional parameters for two-phase flow in small channels. As for hydrodynamics of two-phase flow, a short, but sufficient analysis is presented in [29]. A general form of a similarity equation is as follows

$$F(\mathrm{Eu}, \mathrm{Re}, \mathrm{Bo}, \mathrm{We}, \rho_L/\rho_G, \mu_L/\mu_G) = 0. \tag{9}$$

The nondimensional criteria and simplexes included in the above equation allow to derive another nondimensional numbers, which can be sometimes more convenient for some specific problems, but are not the independent ones. In particular, different combinations of these criteria and simplexes can give Froude number Fr, Kapitsa number Ka, Laplace number Lp, Kutateladze number Ku and so on. For single-phase heat transfer well known nondimensional parameters have to be used for

microchannels at Kn<<1. In flow boiling the mentioned above nondimensional parameter $B = h_{LG}(\nu\rho_G)^{3/2}/(\sigma(\lambda T_S)^{1/2})$ and proposed by Labuntsov nondimensional HTC $Lb=\alpha(\nu\sigma T_S)^{1/3}/(q\lambda)^{2/3}$ are sufficient to reflect the main regularities of the process. One can see that both "new" criteria invented in [15] are easy obtained from the commonly used ones if to use the average rate of vaporization $q/(h_{LG}\rho_G)$ as a characteristic velocity. The parameter B and simplex ρ_L/ρ_G can account for pressure influence on flow boiling regularities in any practically used channels.

TWO-PHASE FLOW IN A BED OF SPHERICAL FUEL ELEMENTS

As it was discussed above, at present no experimental data on flow boiling in a bed of fuel elements are available. In this situation it is reasonable to undertake a preliminary theoretical modelling of the process. Any physical model inevitably is inherently contradictory. Some actual features of a phenomenon are considered as negligible ones in order to understand better the other features, which are assumed to be essential ones. This is true even in relation to such fundamental models as, for example, the model of continuum, but it is more so in relation to any particular model. Undoubtedly, two-phase flow in a round capillary is strong simplification in comparison to real two-phase flow in an isotropic porous media. But what is possible to contrast to this simple model? In the previous section it was discussed that empirical or semiempirical correlations for conventional tubes are not able to predict even integral pressure drop in microchannels, not speaking about flow pattern transitions. Detailed studies of two-phase flow structure in porous media with rather intensive internal heat sources are unknown. Confining the modelling to rather high reduced pressure and to relatively large hydraulic diameter of porous space one gets additional arguments in favour of significance of such analysis.

Two main flow patterns have been considered: an intermittent flow and an annular-dispersed flow. The former incorporates such flow patterns, which could be assumed as homogeneous ones (bubbly, slug and emulsified flow regimes). (It should be pointed out that we consider essentially larger channels and higher pressures, than it was done in [8].) For this flow pattern the modified homogeneous model can be used. In distinction to the standard one a mixture density is calculated in accordance with [29] on the basis of void fraction φ, but not on volumetric flow quality β:

$$\rho_{tp}=\varphi\rho_G + (1-\varphi)\rho_L; \tag{10}$$

$$\varphi = \frac{\beta}{1.1+\varepsilon U_\infty/w_m}. \tag{11}$$

In other words, vapour phase slip in relation to liquid is accounted for. A slip velocity in accordance with the Labuntsov model is determined as a Taylor bubble velocity

$$U_\infty=0.35(g\Delta\rho/\rho)^{1/2}, \tag{12}$$

multiplied by an empirical dimensionless factor

$$\varepsilon=1.4(\rho_L/\rho_G)^{1/5}(1-\rho_G/\rho_L)^5. \tag{13}$$

The latter accounts for vapour bubbles interaction, i.e. "collective effects" in two-phase flow.

Using eqs (10) – (13) one can calculate a density of two-phase mixture ρ_{tp} and then wall shear stress as follows

$$\tau_w=\xi_{th}\,\rho_{tp}w_m^2/8. \tag{14}$$

Friction factor ξ_{th} is computed according to ordinary correlations for single-phase flow. Since in the considered flow pattern the wall surface is always covered with liquid, the liquid viscosity is used for Reynolds number calculation. It means that Reynolds number of two-phase flow is calculated as for single–phase liquid with the entire flow rate of two-phase mixture. At turbulent flow regime the

friction factor is calculated according to eq. (5). In [29] it was demonstrated that this approach essentially improves the standard homogeneous model. A comparison with the experimental data on pressure drop of steam-water in-tube flow at 4.9 and 9.8 MPa (flow quality up to 0.25 and mass flow rate 2000 kg/(m^{2}s)) presented in [29] tests the validity of this approach.

For vertical upward two-phase flow the Taitel model [20] gives good results for transition to the annular flow pattern in conventional channels. In accordance to accepted assumptions and to some mentioned above experimental observations one can anticipate that at high pressures (p/p_{cr}>0.2) two-phase flow in the channels of 1-2 mm in diameter does not differ from flow in the larger channels. On the other hand, in [29] it is shown that at high pressures the Taitel theory predicts unrealistic low void fraction at the point of the transition to the annular flow pattern. In [29] a practical way for predicting this transition is proposed. In general, inequality Ku≥3.1 has to be fulfilled, but an additional condition φ≥0.64 also to be valid. The latter is based on requirement that at annular flow gas (vapour) must be dispersive medium.

In small channels the annular flow patterns were observed either at low [13] and rather high [22, 26] reduced pressures. Commonly modelling of annular-dispersed flow meets two main problems: determining shear stress at the boundary of separated flows of gas and liquid (i) and finding mass flow rate of liquid drops in vapour (gas) core (ii). The latter is obviously less significant for two-phase flow in tortuous channels of porous space of a bed of heated spheres. In [8] it is justly pointed out that due to a channel axis curvature drop deposition has to be very intensive. One can hope that an actual flow at these conditions is close to ideal annular flow. Thus, the only problem of the interface shear stress determination remains.

The well-known Wallis model [30] can be used for predicting friction factor at the liquid film surface;

$$\xi_I = \xi_0(1+k_1\delta/d_h), \tag{15}$$

where k_1 is a numerical factor, which is equal to 300 according to [30]. In [29] in accordance with the later recommendations it is accepted k_1= 240. If the mass flow rate of liquid drops in a vapour core is negligible, one can obtain a closed mathematical description of momentum transfer in the annular flow pattern. All necessary equations are presented in [29]. The aim of solution is dimensionless film thickness δ/d_h, which determines void fraction and pressure drop in two-phase flow in this case. However, one point deserves additional commentary. The interface shear stress according to [29] is determined as follows

$$\tau_i = \xi_i \rho_G (w_G - w_{L,\delta})^2 / 8. \tag{16}$$

Usually $w_G >> w_L$ and solution is essentially simplified. But this commonly applied assumption is not valid for the conditions of rather high pressure and flow velocities. Our approach considers vapour flow in the reference system connected with the liquid film surface. To obtain liquid velocity at the interface $(w_{L,\delta})$ the simple Prandtl model of turbulent flow can be used. With the above assumptions the closed set of algebraic equations in relation to dimensionless liquid film thickness is obtained. Its solution gives values of averaged liquid and vapour velocities and liquid film thickness in any cross section of the channel. This is noteworthy that using for gas flow analysis the reference system moving with a velocity of liquid at the film interface greatly decreases predicted pressure in comparison to the commonly used Wallis model.

The above model has been applied for calculating void fraction, pressure drop and necessary internal parameters of two-phase flow in a bed of spherical fuel elements. A bed of stainless steel spheres of 3 mm in diameter is considered with porosity ε=0.4. Hydraulic diameter of the porous space d_h=2 mm in this case. The height of the bed is assumed to be 250 mm, and the length of the model capillary in accordance to [3] is higher √2 time, that is 354 mm. Induction heating of the metallic balls is assumed. The calculations were performed for intensive enough conditions: heat flux at the heated sphere surface is 1 MW/m^2 (this corresponds to 300 MW/m^3 for the entire bed). Cooling water is supplied from the bottom at saturation temperature at pressure 7.0 MPa; its mass velocity determined from the condition β_{ex}=0.9 (x_{ex}≈0.31) G≈ 1600 kg/(m^{2}s).

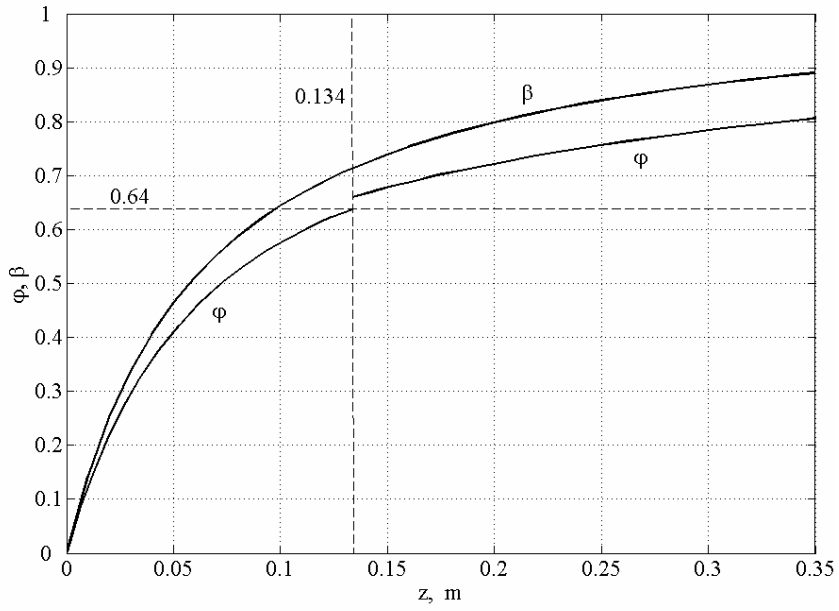

Fig. 1. Volumetric quality and void fraction distribution along the model capillary; p=7.0 MPa, q_w=1 MW/m^2, G =1600 kg/m^{2}s

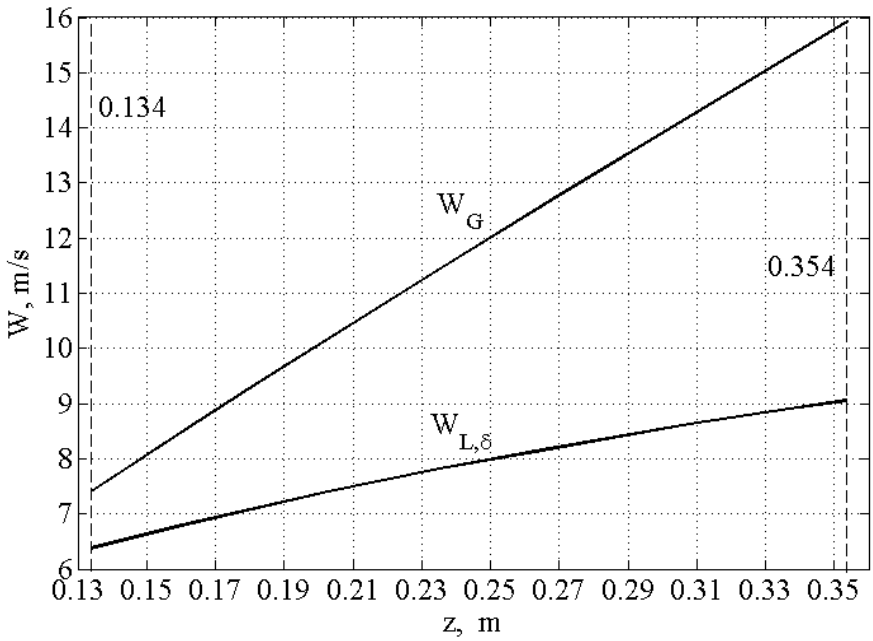

Fig. 2. Vapour and liquid film surface velocities at the interval of annular flow in the model capillary

Fig. 1 depicts the results of calculations for void fraction along the model capillary. As is seen, at **φ**=0.64 flow pattern transition is assumed. This transition is located under the conditions accepted at z=134 mm that corresponds to the height of the bed about 95 mm. The volumetric flow quality at the transition cross section is **β**≈0.73. This indicates a remarkable distinction of the used prediction method from the standard homogeneous model. At the transition cross section the predicted void fraction suffers a small jump. This is natural, because the other interdependence connects the flow parameters at the annular flow pattern. Factually a definite length interval of the channel has to correspond to the region of flow pattern transition. The exit prediction void fraction is $\boldsymbol{\varphi}_{ex}$≈0.808. This means that dimensionless liquid film thickness is equal at the channel exit to 0.05, i.e. liquid film of

0.1 mm in thickness remains under the conditions considered. Evidently, rather large stock to boiling crisis is provided.

Fig. 2 demonstrates the above discussed essential value of liquid velocity at the film surface in the annular flow pattern. This value varies in the range about 6-9 m/s. It is clear that neglect with this velocity leads to great error.

Thus, the calculations give rather optimistic estimations for flow boiling conditions in the bed of fuel microelements of 3 mm in diameter. Total pressure drop of the bed layer of 250 mm in height according to the calculations is also not very large (about 1.5 bar for the accepted heat flux and mass flow rate of coolant). These calculations can be used in design of the experimental facility. There is no doubt that in two-phase flow only experimental verification allows final estimating a value of any model.

Nomenclature

A – nondimensional bubble growth module
$B = h_{LG}(\nu\rho_G)^{3/2}/(\sigma(\lambda T_S)^{1/2})$ – dimensionless parameter
$b = (\sigma/(g\Delta\rho))^{1/2}$ –Laplace constant, m
Bo – Bond number, dimensionless
c_p – liquid specific heat, J/(kg K)
D –bubble diameter, m
d_h – hydraulic diameter of channel, m
Eu – Euler number, dimensionless
g – gravitational acceleration, m/s^2
h_{LG} – latent heat of evaporation, J/kg
$Ja = (\rho_L c_p \Delta T)/(h_{LG}\rho_G)$ – Jakob number, dimensionless
p – pressure, Pa
Pe – Peclet number, dimensionless
Pr – Prandtl number, dimensionless
q – heat flux density, W/m^2
R – bubble radius, m
Re – Reynolds number, dimensionless
r_0 – channel radius, m
T – temperature, K
t – time, s
U_∞ – terminal velocity of vapour slug (Taylor bubble), m/s
u, u_0 – flow velocity, mean flow velocity, m/s
We – Weber number, dimensionless
w_m – mixture velocity, m/s
x – flow quality, dimensionless
y – distance from the wall surface, m
α – heat transfer coefficient, W/(m^2 K)
β – volumetric flow quality, dimensionless
δ – liquid film thickness, m
φ – void fraction, dimensionless
κ – liquid thermal diffusivity, m^2/s
λ – liquid thermal conductivity, W/(m·K)
μ – liquid dynamic viscosity, kg/(ms)
ν – liquid kinematic viscosity, m^2/s
ρ – density, kg/m^3, $\Delta\rho = \rho_L - \rho_G$
τ – shear stress, Pa

Subscripts

cr – critical	d – departure
ex – exit	G – vapour (gas)
i – interface	L – liquid
s – saturation	tp – two-phase
w – wall	

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