

## MECHANISM OF TRANSIENT LIQUID BOILING

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*An analysis is made of current views on transient liquid boiling. A physical model is delineated, based on which characteristic times of the process and heat transfer are predicted.*

In the late 50s the author of [1] complained that transient boiling rarely draws the attention of researchers. Judging from the number of reports submitted to the 9th International Heat Transfer Conference [2-6], over 35 years the situation has not changed radically. And this is contrary to the fact that in recent years interest in transient boiling has been fostered by analyzing the problems of safety of nuclear reactors, since in a hypothetical emergency with coolant loss consideration is inevitably given to active-zone cooling in a transient boiling mode [2].

The author of [1] had at his disposal only scanty data on heat transfer in transient boiling and results of process filming. Today we have not only voluminous experimental data on heat transfer but also information important for gaining insight into the process mechanism, as to the times of contact of the liquid and the vapor with the wall [7, 8] and as to the dependence of the fraction of the wetted heating surface on the surface temperature. Still, even the latest publications note that transient boiling is the least understood type of boiling [2, 3] and that there is neither understanding of the main mechanism of the process nor dependable calculating relations for heat transfer in transient boiling [6].

A fairly detailed analysis of present views on transient boiling is contained in [2, 8]. According to [2], most of the models of heat transfer in transient boiling are based on the conclusion [12] that transient boiling is a combination of nonsteady modes of nucleate and film boiling, each of them existing alternately at a given spot of the heating surface. The general structure of the expression for the average heat flux density in transient boiling, following from such models, appears as

$$q = \gamma q_N + (1 - \gamma) q_F, \quad (1)$$

where  $q_N$  and  $q_F$  are the average, for the given conditions, heat flux densities for nucleate and film boiling, respectively. The quantity  $\gamma$  is the fraction of the area of the heating surface whereon wall-liquid contact, i.e., nucleate boiling, occurs. For an ergodic process (for rather extended heating surfaces, the ergodicity condition is fulfilled [7])

$$\gamma = F_l/F \approx t_l/t_t, \quad (2)$$

that is, the fraction of the surface area occupied by the liquid ( $F_l/F$ ) is equal to the fraction of time of liquid-wall contact ( $t_l/t_t$ ) at the given point of the surface, if the total observation time  $t_t$  is much longer than the characteristic time of the process (the period of liquid-wall contacts).

The values of  $q_N$  and  $q_F$  in Eq. (1) are expressed differently in different works. According to one of the commonly used approaches (see [2]),  $q_N = q_{cr}$  and  $q_F = q_{min}$ , where  $q_{cr}$  and  $q_{min}$  are the first and second critical heat flux densities for the liquid considered at the corresponding pressure. In this case,  $\gamma = 1$  at  $q = q_{cr}$ , and  $\gamma = 0$  at  $q = q_{min}$ . Today there are no relations  $\gamma(\Delta T)$  in the interval from  $\Delta T_{cr}$  to  $\Delta T_{min}$  that are at all reliable and physically justifiable. The attempts to utilize the results obtained in [4, 7-9, 11] from direct experimental measurements of this relation can at most (in the case of success) verify the expediency of using equations of the type (1), but they can hardly be regarded as a physical model of the process. Study [13] demonstrated that, firstly, with  $q = q_{cr}$  the fraction of the wetted surface is noticeably smaller than unity and, secondly, the relation  $q(\gamma)$  per se is essentially nonlinear, which is contrary to Eq. (1). In [13], for analyzing the proper data the quantity  $q_N$  in Eq. (1) is defined by the

empirical nonlinear (cubic) equation with respect to the dimensionless temperature

$$\theta = \frac{T_w - T_{cr}}{T_{min} - T_{cr}}.$$

Such a technique is, in essence, similar to approaches where the nonlinear function  $\theta$  was selected to describe  $\gamma$  variations in the transient boiling region [14].

Much the same is a scheme [15] according to which an equation of the form (1) is written for the heat transfer coefficient  $\alpha$  in transient boiling. Here the authors presume  $\alpha_N \gamma$  to be a single-valued function of  $\Delta T = T_w - T_s$  ( $T_s$  is the saturation temperature of the liquid) for the given surface-liquid combination. Study [15] also suggests to introduce a correction factor for  $\gamma$  that takes into account the surface wettability and the direction of the heat flux change (heating or cooling).

The strong influence of the surface properties on the transient boiling characteristics is beyond question [2-4, 10, 15-17]. A quantitative reflection of this influence in calculating relations for heat transfer appears rather hopeless, especially as the wetting angle is a characteristic very poorly reproducible even under the conditions of a "pure" physical experiment. For this reason, apparently, we will have to be reconciled to the fact that any calculating relation for the heat transfer coefficient in transient boiling can only describe a certain average level of the process rate, and differences in the quantities predicted or measured under some or other specific conditions may be quite appreciable. An alternative to such an approach involves, obviously, only direct experimental modeling of the processes proceeding in the units being designed, which is hardly possible in all cases.

Representative of the state of developing the calculating relations in the transient boiling region is study [6], reporting an experimental investigation for water transient boiling, performed very thoroughly using precision sensors and up-to-date secondary instruments. (Suffice it to say that the temperature was measured by thermocouples with a 25  $\mu\text{m}$  o.d. cable, flush-mounted with the heat transfer surface). Excellent reproducibility and quite insignificant spread in the experimental points were achieved in the experiments [6]. However, the deviation of the points from the empirical calculating relation selected by the authors of [6], especially for describing the proper experimental data, appeared to be markedly large (84% of the points in the band of  $\pm 20\%$ ).

Studies where the calculating relations for heat transfer are derived from a equation of the form (1) do not, as a rule, make attempts to link these relations with a detailed analysis of the transient boiling mechanism. The same applies also to study [18], which suggests to obtain the boiling curve (including the section of transient boiling) based on its formal analogy with the Poisson distribution. In distinction to such approaches, the authors of the [8] and (especially) of [19] worked out quite a comprehensive transient boiling model. Study [8] (and, previously, [20]) derived a semiempirical relation for the heat transfer coefficient in transient boiling, taking into account constituents of the heat flux resulting from nonsteady cooling of the wall in the beginning of its contact with the liquid, as well as from nucleate and film boiling. Characteristic times of each of the three stages of the process are determined from model representations up to constant factors. It is suggested to calculate the heat flux constituents that account for the periods of liquid evaporation and wall-vapor contact on the basis of extrapolation of the appropriate relations  $q(\Delta T)$  for nucleate and film boiling to the transient boiling region.

Study [13] and, later, [2] give sufficiently justified objections to using the relation  $q(\Delta T)$ , extrapolated from the transient boiling mode, to describe the transient region. Actually, direct experiments [13, 2] indicate that the heat flux density  $q_N$  during wall-liquid contact in the transient mode decreases with increasing  $T$ , whereas [8] suggests, in fact, the relation  $q_N \sim \Delta T^3$ , predicting a very rapid rise of  $q_N$  with increasing wall superheating. In [21], we cast one more doubt on the validity of the model [8] because the experimental times of liquid-wall contact in transient boiling amount to a few milliseconds, which is smaller than the characteristic times of the periodic processes of nucleation and of bubble growth and detachment in nucleate boiling (this is the case both with individual bubbles and with vapor agglomerations, typical of high values of  $q$ ). The resultant relation of study [8] for the boiling of a saturated liquid contains three free constants (and six for a subcooled liquid) and, besides, involves an indeterminate, in practical calculations, quantity such as the wetting angle. Obviously, studies [8, 20] fall far short of resolving the problem of working out a model of heat transfer in transient boiling.

Study [19] appears quite substantiated physically in the part with a qualitative description of the stages of

transient boiling. However, in in-depth analysis of individual stages of the process the authors employ approaches some of which provoke decisive objections. A comprehensive treatment of these approaches might make up the content of a separate publication, but the most crucial objection is caused by the use of the effective turbulent thermal conductivity of the liquid  $\lambda_{\text{eff}}$  to describe nonsteady wall cooling in liquid-wall contact. Judging by the fact that the authors of [19] begin the conclusions of their article by asserting the importance of taking into account liquid turbulence in analyzing transient boiling, they consider such an account a significant advance. Meanwhile, it is well known that, even at very large Reynolds numbers of the main flow, molecular momentum and energy transfer is always prevalent at the wall. Over the times of nonsteady cooling of the order of  $10^{-6}$  sec stated in [19], the region of the temperature disturbance propagates into the liquid for distances shorter than  $10^{-6}$  m (even if we resort to the values  $\lambda_{\text{eff}} \approx 100\lambda$  given in [19], the depth of the temperature disturbance is not in excess of  $10 \mu\text{m}$ ). Evidently, the indicated distances lie within a viscous sublayer even at very large Re numbers. With the expressions for the average rate of turbulent pulsations and the linear scale utilized in [19], the thickness of the viscous sublayer for water should be as large as  $100 \mu\text{m}$ .

The authors of [19] relate for some reason a sharp fall of the wall temperature precisely to the interaction between the bulk of the cold liquid and the superheated wall (temperature fluctuations of the wall in transient boiling may exceed 100 K). Meanwhile, as early as 30 years ago it was proved convincingly that a sharp fall of the heating surface temperature may be associated only with evaporation of a thin liquid layer (a "microlayer"). The authors of [19] use in their model the notion of a large vapor agglomeration with an underlying thin liquid layer (a "macrolayer"). In that case they calculate the heat flux density in the period of macrolayer evaporation from the Rossenou equation for heat transfer in nucleate boiling. Although such an approach is doubted by the authors of [19] themselves, even having agreed to it, it is impossible to understand why, at the instant of liquid-wall contact, they take into account wall cooling and, in the evaporation of the liquid macrolayer, they regard the wall as isothermal despite the fact that the heat flux densities in the latter case are 1-2 orders of magnitude higher (according to their own evaluations). However, with the macrolayer thickness known from experimental measurements and calculations [19] (of the order of  $10^{-5}$  m), the heat flux densities during macrolayer evaporation in water boiling must be even higher (several  $\text{MW}/\text{m}^2$ ).

Fitting of results calculated from the model [19] to experimental data [7] is achieved by matching the coefficients. These calculations are in poor agreement with experiments [9, 11], although study [19] has failed to obtain the necessary relations (for the heat transfer coefficient and the time of liquid-wall contact) in the form of explicit analytic equations. The desired relations result only from numerical solution.

On this background, a transient boiling model delineated in [21] appears to be a step forward. Qualitatively, it resembles the model of [19]. Consideration is given to three sequential stages of transient boiling. After the departure of the vapor volume on the surface section under examination the liquid comes in contact with the solid surface. At the first stage, superheating of the liquid layer with a thickness of the order of a few radii of a viable nucleus  $R^*$ , boiling up, and primary growth of nuclei occur until they collapse and form a vapor agglomeration. This stage is 2-3 orders of magnitude shorter than the two subsequent stages, and its role in heat transfer may be disregarded. Such a deduction is qualitatively consistent with results of [19], but here, of course, the turbulent thermal conductivity of the liquid is not introduced artificially in order to explain the fall of the wall temperature. The temperature fall should occur at the second stage, during which a liquid film at the base of the vapor agglomeration evaporates completely. The model employs an isothermal wall approximation, i.e., the wall temperature in boiling is assumed to remain constant, equal to the average wall temperature  $T_w$ , measured experimentally. Here, the total amount of heat removed from the wall during evaporation is specified by the thickness of the liquid film, and the evaporation time, as opposed to [19], is defined by the law of growth of the vapor volume. At the end of the second stage, liquid-wall contact ceases. The time of the third stage, during which the wall contacts the vapor, is determined by the conditions of detachment (of rejection by the liquid) of a large vapor bubble. Thus, whereas qualitatively our model deals with the same basic processes making up the transient boiling cycle as [19], their detailed content and characteristic quantitative regularities are entirely different.

Consider more specifically each of the three stage of the process. In order for the liquid contacting the hot wall to boil up, it is sufficient to ensure its superheating in a layer of thickness somewhat in excess of the radius of

viable nucleus  $R_*$ , since, in the preceding stage, the superheated solid surface touched the vapor and, therefore, all its cavities (potential evaporation centers) at the instant of coming in contact with the liquid were filled up with the vapor. The size of a viable nucleus is defined by the relation

$$R_* = 2\sigma/\Delta p, \quad (3)$$

where  $\sigma$  is the surface tension; and  $\Delta p = p'' - p_s$  is the pressure jump at the interface, i.e., the difference in the vapor pressure between the bubble and the liquid (it is logical to assume the interfacial pressure from the side of the liquid to be equal to that above the liquid level, if the latter is not too low). Because transient boiling proceeds at large  $\Delta T$  ( $\Delta T > \Delta T_{cr}$ ), the characteristic values of  $R_*$  are very small here ( $10^{-7}$ - $10^{-6}$  m). The heating time for a liquid layer of thickness even an order of magnitude larger is not longer than  $10^{-6}$  sec. Thus, within  $10^{-6}$  sec of the liquid coming in contact with the wall, vapor bubbles start growing.

The initial stage of growth, obviously, obeys the Rayleigh law [24]

$$R = \sqrt{\frac{2}{3} \frac{\Delta p}{\rho'}} t, \quad (4)$$

where  $\rho'$  - is the liquid density. A value of  $R \approx 10R_*$  is attained over a time  $t < 10^{-7}$  sec for the parameters corresponding to the region of transient boiling for most liquids. The density of the evaporation centers may be assessed from the scheme suggested in [25]:

$$n_F = C_1 R_*^{-2}.$$

For nucleate boiling,  $C_1 \approx 10^{-8}$ - $10^{-7}$ . It is believed that this constant is 2-3 orders of magnitude larger in transient boiling. Therefore, the average distance between the boiling centers is

$$l = \sqrt{1/n_F} = C_2 R_* \quad (5)$$

with the constant  $C_2 = 10$ - $100$ . With such "close packing" of the boiling centers, vapor bubbles growing therein by law (4) will merge in the middle part over a time of  $10^{-7}$ - $10^{-6}$  sec. Therewith a vapor agglomeration (a "vapor mushroom") forms, at whose base the liquid film persists with a thickness of the same order of magnitude as  $l$ , i.e.,

$$\delta = C_2 R_*. \quad (6)$$

Figure 1 gives a sketch of this stage of the process.

Since the characteristic times of liquid layer heating, boiling up, and initial growth of vapor bubbles with generation of a vapor agglomeration are about three orders of magnitude smaller than the time intervals characterizing other stages, this stage can be assumed to proceed "instantaneously." This circumstance has, apparently, led the author of [1] to the conclusion, based on analyzing the results of high-speed filming of the process, that in transient boiling the liquid does not touch the wall but only approaches it very closely, gets superheated, and evaporates explosively. Clearly, times of the order of  $10^{-6}$  sec are not observable at a filming speed of the order of 1000 frames per second. However, the process of explosive boiling up of the liquid that has approached the wall, described qualitatively in [1], agrees quite well with the advanced physical model.

In the second stage, representing evaporation of a thin liquid film, the major part of the heat is removed from the wall. The duration of this stage (i.e., the time of liquid-wall contact  $t_l$ ) is governed by the film thickness and the evaporation rate, which, in turn, is related directly to the rate of growth of the vapor agglomeration volume. At large Jacob numbers  $Ja$ , according to [26], the rate of the vapor bubble growth on the heated solid surface is

$$\dot{R} \equiv \frac{dR}{dt} = \frac{3}{8} \left( \frac{\lambda c_p}{\rho''} \right)^{1/4} \frac{\tilde{R}^{3/4} T_s^{5/4}}{r} t^{-1/4}, \quad (7)$$

where  $\lambda$ ,  $c_p$ , and  $r$  are the thermal conductivity, specific heat, and evaporation heat of the liquid;  $\rho''$  is the density of the saturated vapor at  $T_s$ ; and  $\tilde{R}$  is the gas constant (individual). Equation (7) is validated for  $Ja \geq 500$  ( $Ja = \rho' c_p \Delta T / (r \rho'')$ ), i.e., for large wall superheatings and relatively low pressures when the bubble growth is strongly affected by inertial processes, which causes a rise of the pressure on the outer border of the bubble (of the vapor

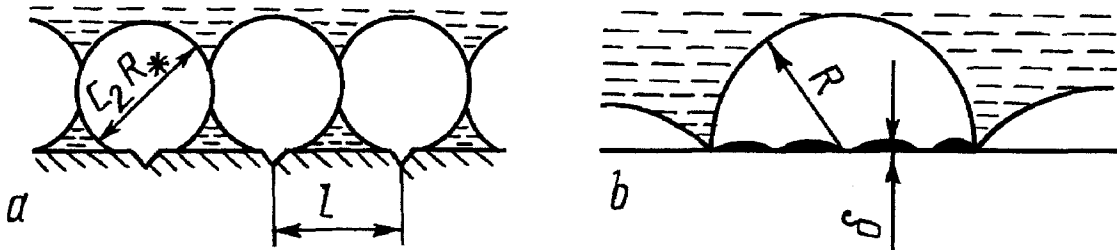


Fig. 1. Sketch of collapse of vapor bubbles in liquid boiling up (a) and in subsequent growth of the vapor volume (b).

pressure in the bubble) in comparison with  $p_s$ . These effects are also considerable at smaller Ja numbers at the initial stage of bubble growth. The times of liquid-wall contact, measured in experiments [7, 8], were not longer than  $10^{-2}$  sec, which constitutes a small part of the time of bubble growth on the surface at relatively large Ja numbers. This gives grounds to believe that law (7) is applicable to transient boiling conditions also at noticeably smaller (than 500) Ja numbers. By our evaluations [21], the applicability limit of law (7) is determined by the inequality  $Ja \geq 200$  (for water under atmospheric pressure, this conforms to the condition  $\Delta T \geq 67$  K).

Let us assume that vapor agglomeration grows by law (7) "from a point" (i.e.,  $R = 0$  at  $t = 0$ ), retaining therewith a hemispherical shape (see Fig. 1), up to the instant of complete evaporation of the liquid film  $t_l$ , when the bubble radius  $R_l$ , final for this stage, is defined by the material balance

$$\frac{3}{2} \pi R_l^3 \rho'' = \pi R_l^2 \delta \rho' = \pi R_l^2 C_2 R_* \rho'.$$

Then

$$R_l = C_2' R_* \rho' / \rho'', \quad (8)$$

where the constant  $C_2'$  is of the same order of magnitude as  $C_2$ .

From Eqs. (7) and (8) we obtain the time of complete evaporation of the liquid film:

$$t_l = \frac{(2C_2')^{4/3} (R_* \rho' r)^{4/3}}{(\lambda c_p)^{1/3} \tilde{R} \rho'' T_s^{5/3}}. \quad (9)$$

The pressure drop  $\Delta p$  along the saturation line in Eq. (3) for large temperature drops  $\Delta T$  cannot be expressed by the linear relation derived from the Clapeyron-Clausius equation [21, 26, 27]. For  $\Delta T$  of the order of  $0.1 T_s$ , in conformity with [26], we have

$$\Delta p \approx r^2 \rho'' \Delta T^2 / (\tilde{R} T_s^3).$$

Using this relation in Eq. (3) and the value of  $R^*$  obtained in Eq. (9), we find

$$t_l = C_3 \left( \frac{\rho' \sigma}{r} \right)^{4/3} \left( \frac{\tilde{R}}{\lambda c_p} \right)^{1/3} \left( \frac{T_s}{\rho''} \right)^{7/3} \frac{1}{(\Delta T)^{8/3}}. \quad (10)$$

The constant  $C_3 = (2C_2')^{4/3}$  should be of the order of  $10^2 - 10^3$ . As is evident from Fig. 2, Eq. (10) at  $C_3 = 500$  shows reasonable agreement with experimental data [7, 8]. According to the analysis of [2], the study [7] obtained more reliable data on the times of liquid-wall contact. The spread of experimental points that reflects the statistical nature of the process is very large, on the whole, the experimental data fit the prediction from Eq. (10) not only qualitatively but also quantitatively.

The completion of liquid film evaporation is followed by the third stage, during which the liquid is separated from the solid surface by the vapor. In accordance with [19], at this stage relatively slow growth of the vapor volume occurs due to the heat supplied to the liquid from the heating surface through a vapor film. Here, an inertial approximation is employed for detachment of the vapor bubble from the wall. While the second assumption does not cause any objections, the first is highly questionable. Indeed, the heat flux from the wall is calculated in [19], in the

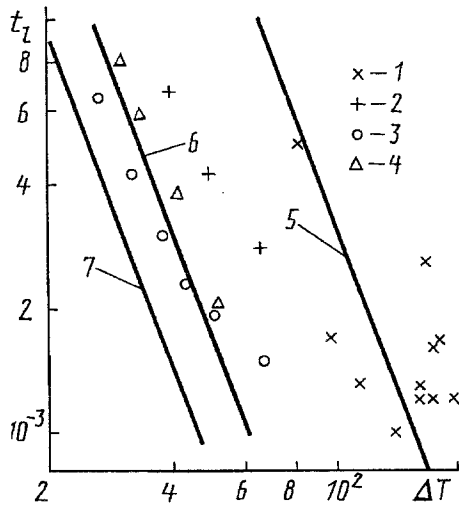


Fig. 2. Time of liquid-wall contact  $t_l$  vs temperature head  $\Delta T$ : 1) water, 0.1 MPa [7]; 2) water, 0.1 MPa [8]; 3) ethanol, 0.1 MPa [8]; 4) Freon-113, 0.1 MPa [8]; 5, 6) prediction from Eq. (10) for water, ethanol, and freon-113, respectively.  $t_l$ , sec;  $\Delta T$ , K.

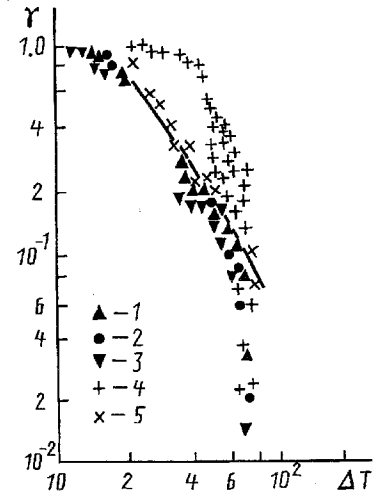


Fig. 3. Fraction of the wetted heating surface vs temperature head in transient boiling of methanol under atmospheric pressure: 1-3) results of three measurement runs in a steady mode [13]; 4, 5) measurement results for a nonsteady cooling mode in studies [9] and [11], respectively; the curve shows calculation from Eqs. (15) and (16).

mode under consideration, from relations for film boiling. However, in the film mode, the heated wall is covered with a fairly thin vapor film, whereas, in conformity with the model adopted in [19], at the end of the period of liquid "macrofilm" evaporation, in the considered wall region on the contrary, there is a very sizeable vapor volume (which also has a large dimension in the direction normal to the solid surface). This is shown by frames of high-speed filming of the process, presented in [1], and by a qualitative description of the process in [20].

It appears more justifiable to assume that further expansion of the vapor bubble volume on completion of liquid film evaporation is due to excess vapor pressure in it, produced during its rapid growth at the preceding stage. At least, such a hypothesis is quite realistic for low pressures, for which the condition  $Ja \geq 200$  is fulfilled. An approximate estimation of the excess pressure at the instant  $t_l$ , i.e., at the end of the evaporation period of the liquid film, gives

$$\Delta p_l'' \approx \frac{3}{2} \rho' \dot{R}_l^2, \quad (11)$$

where  $R_l$  is found from Eq. (7) with allowance for Eq. (10). It can readily be shown readily that

$$\Delta p_l'' = C_4 (\rho')^{1/3} \left( \frac{\rho'' \lambda c_p}{\sigma} \right)^{2/3} \left( \frac{\bar{R} T_s \Delta T}{r} \right)^{4/3}, \quad (11')$$

where  $C_4 \approx 3/2C_3^{-1/2} \approx 0.07$ . The calculation from Eq. (11) for water under atmospheric pressure produces the very perceptible value  $\Delta p_l'' \approx 1.2 \cdot 10^4$  Pa at  $\Delta T = 100$  K.

The excess energy in the vapor cavity, caused by the pressure  $\Delta p_l''$ , at the instant it starts to expand freely (on completion of an intense vapor inflow to the cavity) may be evaluated as

$$E_0 = \Delta p_l'' V_l'' = \frac{4}{3} \pi R_l^3 \Delta p_l''.$$

Assuming the cavity to expand by the law that determines the change of the radius of a hemispherical cavity in

underwater explosion [28], we obtain

$$R \left[ 1 - \left( \frac{R_l}{R} \right)^{5/2} \right]^{2/5} = \text{const} \left( \frac{\Delta p_l'' R_l^3}{\rho'} \right)^{1/5} t^{2/5}. \quad (12)$$

The value of const should be of the order of 1.0.

Large bubbles get detached (depart from the wall) by the inertial scheme of [29]. Then, at the detachment instant  $t_d$

$$R_d \approx 0,5gt_d^2, \quad (13)$$

where  $g$  is the acceleration of mass forces (under "normal" terrestrial conditions, of the gravity force).

If  $R_d \gg R_l$  at the instant of detachment of the vapor volume, the initial condition slightly influences the time of growth up to  $R_d$ . In this case, on the left-hand side of Eq. (12), the factor at  $R$  can be equated to 1.0. Then, eliminating  $R = R_d$  from Eqs. (12) and (13), we find the time for vapor volume expansion at the wall, i.e., the time of vapor-wall contact  $t_v = t_d$ . Taking into account the values of  $R_l$  from Eq. (8) and of  $\Delta p_l''$  from Eq. (11), we obtain

$$t_v = C_5 \frac{(\rho' \sigma)^{7/24} (\lambda c_p)^{1/12} \bar{R}^{13/24} T_s^{31/24}}{(\rho'')^{2/3} r^{11/12} g^{5/8} \Delta T^{7/12}}. \quad (14)$$

The constant  $C_5$ , as follows from the derivation of Eq. (14), should be of the order of unity.

The time fraction of liquid-wall contact is

$$\gamma = t_l/t_t = (t_l/t_v)/(1 + t_l/t_v), \quad (15)$$

where

$$\frac{t_l}{t_v} = 500 \frac{(\rho' \sigma T_s)^{25/24} g^{5/8}}{(\lambda c_p r)^{5/12} (\rho'')^{5/3} \bar{R}^{5/24} \Delta T^{25/12}}, \quad (16)$$

as follows from Eqs. (10) and (14) at  $C_5 = 1$ .

Figure 3 compares the relation  $\gamma(\Delta T)$ , calculated from Eqs. (15) and (16), and experimental values of the fraction of the wetted area  $F_l/F$  in transient boiling of methanol under atmospheric pressure [9, 11, 13]. In these studies, boiling occurred on an upward-facing horizontal end surface of an aluminum cylinder, exposed to anodizing. Studies [9] and [11] obtained the boiling curve and values of  $F_l/F$  during nonsteady surface cooling, and [13], in steady transient boiling. The agreement between the calculated curve and the results of stationary (three measurement runs in [13]) and nonstationary experiments [11] is surprisingly good for so intricate a process. Experimental values of  $F_l/F$  obtained in [9] lie noticeably higher than the calculated curve. Study [21] demonstrated that the calculated relation  $\gamma(\Delta T)$  for water under atmospheric pressure, constructed according to Eqs. (15) and (16), passes through the lower part of the array of experimental values of  $F_l/F$  obtained in [9] under nonsteady conditions (cooling).

The appreciable distinction in the position of boiling curves for steady and nonsteady conditions was noted in [15, 2, 3] and is quite clear from comparing the results of studies carried out with different directions of change in the heat flux [2, 4, 15-17]. The general scantiness of experimental investigations of transient boiling does not, unfortunately, makes it possible, when comparing calculated relations with experimental results, to select numerical constants based only on measurements performed under identical conditions. For example, reliable data on the wetted surface fraction in transient boiling in a steady mode are acquired, as far as the author knows, only in [13]. Under such conditions, it is impossible to require that the calculation relation for both internal and integral characteristics of transient boiling describe equally well the experimental data obtained by different experimental methods. Nonetheless, there are grounds for regarding Eqs. (10) and (14)-(16) as consistent not only qualitatively but also quantitatively with available data on the internal characteristics of transient boiling.

In calculating heat transfer, it is possible to neglect the amount of heat removed during vapor-wall contact, bearing in mind an unavoidable approximateness of any relation for transient boiling. Then, the average heat flux density is likely to be equal to the heat flux density for the time of liquid-wall contact based on the total period of

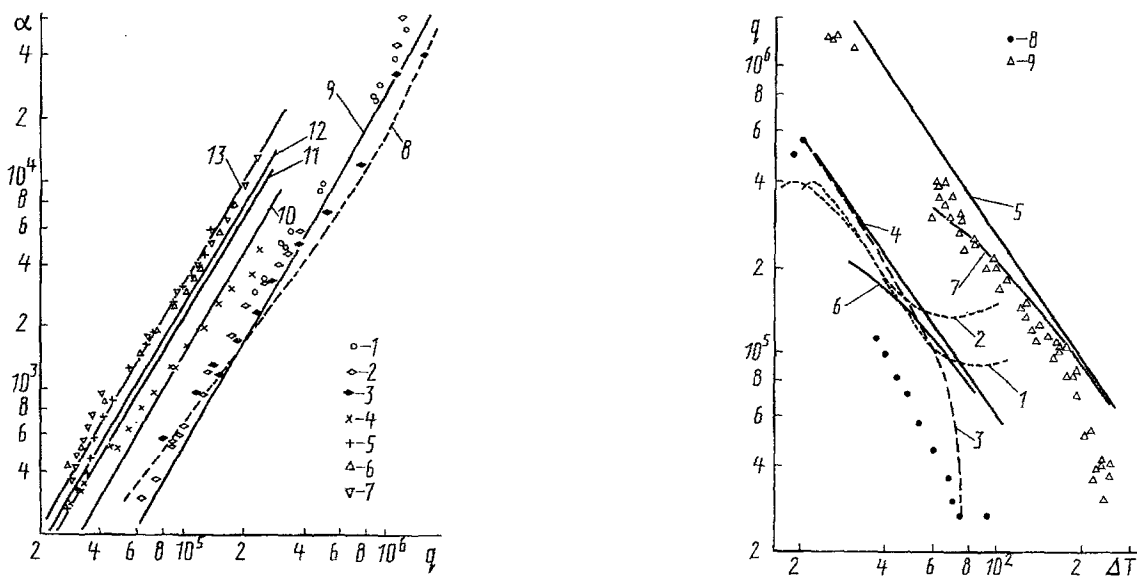


Fig. 4. Heat transfer coefficient in transient boiling of liquids: 1) water, 0.1 MPa [1]; 2) water, 0.1 MPa [30]; 3) water, 0.1 MPa,  $\rho_w = 136 \text{ kg}/(\text{m}^2 \cdot \text{sec})$  (data of S. C. Cheng et al., according to [21]); 4) methanol, 0.1 MPa [1]; 5) freon-113, 0.05 MPa; 6) freon-113, 0.1 MPa; 7) freon-114, 0.3 MPa [31]; 8) calculation from an empirical equation of [6] for water at 0.1 MPa; 9-13) calculation from the present author's equation, correspondingly, for water, methanol at  $p_s = 0.05$  and 0.1 MPa, and freon-114 at  $p_s = 0.3$  MPa.  $\alpha$ ,  $\text{W}/(\text{m}^2 \cdot \text{K})$ ;  $q$ ,  $\text{W}/\text{m}^2$ .

Fig. 5. Heat transfer for methanol and water under atmospheric pressure: 1, 2) smoothed curves of two modes of nonsteady surface cooling in methanol boiling [30]; 3) calculation from Eq. (1) for steady boiling of methanol using experimental values of  $F_l/F$  [13]; 4, 5) calculation from Eq. (17) for methanol and water at  $t_l/t_v = 0$ ; 6, 7) the same but using values of  $t_l/t_v$  according to Eq. (16); 8, 9) experimental points for methanol [13] and water [30] under the conditions of steady transient boiling.

the process  $t_t = t_l + t_v$ :

$$q = \frac{\rho' \delta r}{t_l + t_v} = \frac{C_2 R_* \rho' r}{t_v (1 + t_l/t_v)}$$

Using the values of  $t_v$  and  $R_*$  obtained above, we find

$$q = C_6 \frac{(\sigma \rho')^{17/24} \tilde{R}^{11/24} T_s^{41/24} g^{5/8}}{(r \lambda c_p)^{1/12} (\rho'')^{1/3} \Delta T^{17/12} (1 + t_l/t_v)} \quad (17)$$

The constant  $C_6$  is of the order of 10-100. Study [21], having set  $t_l/t_v \ll 1$ , i.e., having equated the last cofactor in the denominator of Eq. (17) to unity, has assumed  $C_6 = 32$ .

Figure 4 compares calculations from Eq. (17) and experimental data on heat transfer in transient boiling of water [1, 2, 30], methanol [1], and freons-113 and 114 [31]. The form of representation and the experimental data pertaining to heat transfer in transient boiling of water under atmospheric pressure under the conditions of forced motion at a mass velocity of  $136 \text{ kg}/(\text{m}^2 \cdot \text{sec})$  (S. C. Cheng et al., 1978) are borrowed from [2]. The experimental data on heat transfer in transient pool boiling of water under atmospheric pressure, borrowed from [1], are obtained by Braunlich. It is clear from analysis of the figure that, at low pressures, calculations from Eq. (17) are in reasonable agreement with experimental data. Study [21] gives other examples of comparing the calculated relation (17) with experimental data, in particular, with results obtained in [32] for transient boiling of water, hexane, heptane, acetone, and isopropanol at pressures ranging from 3 kPa to atmospheric. Exactly these data were first employed for selecting the value of the constant  $C_6$  in Eq. (17).



Of interest is Fig. 5, reproducing experimental data of studies [13] and [30] on methanol and water boiling under atmospheric pressure. Experimental points [13], characterizing heat transfer in methanol transient boiling in a steady mode, lie appreciably lower than curves 1 and 2, obtained in [30] using an identical setup under the conditions of nonsteady surface cooling. From the standpoint of the above analysis of relation (1), it is very significant that curve 3 in Fig. 5, which is constructed based on this relation with the use of experimental values of  $\gamma = F_l/F$ , obtained in the same experiment [13], and passes through the points  $q_{cr}$  and  $q_{min}$ , deviates considerably from experimental values of  $q(\Delta T)$ . The calculating equation (17) for  $1 + t_l/t_v \approx 1.0$  is in favorable agreement with the experimental data on heat transfer in transient boiling of water under steady conditions [30] but differs noticeably from similar data [13] for methanol, although it is quite consistent with the results of nonsteady experiments with methanol. Curves 6 and 7 in Fig. 5 are plotted with regard to real values of  $t_l/t_v$  according to Eq. (16). Evidently, at least for water (over the range  $\Delta T = 60-200$  K), taking account of this factor improves the agreement with experimental data. At the same time, with small  $\Delta T$ , i.e., as the point  $(q_{cr}, \Delta T_{cr})$  is approached, the value of  $t_l/t_v$  becomes so large that the main model assumptions are not fulfilled. For example, for water at  $\Delta T = 30$  K,  $t_l \approx 70$  msec, in conformity with Eq. (10). Clearly, with such times, the process mechanism is other than that assumed in the model; here it is already possible to speak of coexistence of zones of nucleate and film boiling, as is adopted in [8, 20]. It should be pointed out that the transient boiling section adjacent to a maximum point on the boiling curve is not reproduced in stationary experiments [6, 13, 30], and in the nonsteady cooling experiments it is passed extremely quickly because of very large heat transfer coefficients.

Undeniably, the delineated physical model is highly approximate. Yet, with its help an analytical equation is derived to calculate the transient boiling section without resorting to direct experimental data on the coordinates of the points corresponding to the boiling crisis. Here, relation (17) involves only one empirical constant (under the approximation  $t_l \ll t_v$ , using which the indicated relation was compared with experimental data). The physical premises employed in the current study are valid at relatively low pressures ( $Ja \geq 200$ ). Later on, without discarding the basic points of the present model, we expect to succeed in describing the processes proceeding in the region adjacent to the boiling crisis point as well as the specific features of high reduced pressures.

## NOTATION

$c_p$ , specific heat at constant pressure;  $E_0$ , energy;  $F$ , area;  $F_l$ , wetted area;  $g$ , acceleration of mass forces;  $n_F$ , density of evaporation centers;  $p$ , pressure;  $q$ , heat flux density;  $R$ , bubble radius;  $R_*$ , critical bubble radius;  $\tilde{R}$ , gas constant;  $r$ , heat of evaporation;  $t$ , time;  $T$ , temperature;  $\Delta T = T - T_s$ , where  $T_s$  is the saturation temperature;  $\gamma$ , fraction of the wetted area;  $\delta$ , film thickness;  $\lambda$ , thermal conductivity;  $\rho$ ,  $\rho''$ , liquid and vapor densities;  $\sigma$ , surface tension;  $Ja$ , Jacob number;  $Re$ , Reynolds number. Subscripts:  $N$ , nucleate boiling;  $F$ , film boiling;  $l$  and  $v$ , the liquid and vapor, respectively;  $d$ , the detachment instant;  $t$ , the total time of the cycle.

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