Journal of Engineering Physics and Thermophysics, Vol. 69, No. 5, 1996

CALCULATION OF EVAPORATION OF A TWISTED FLOW OF CRYOGENIC FLUID IN LEIDENFROST FILM BOILING WITH ALLOWANCE FOR THE INITIAL SECTION

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UDC 536.2:621.59

A mathematical model of evaporation of a twisted flow if a cryogenic fluid developed by the authors is supplemented by the calculation of an initial section. Calculational and experimental results are compared.

A mathematical model of boiling of a cryoagent twisted flow that is intended for solving problems of optimization of the design parameters of evaporators has been developed by the authors and is described in [1-3] considered the given process as boiling in the Leidenfrost regime assuming it to be stable starting from the initial section. To solve the problem of optimization one needs computational relations that make it possible to determine heat transfer in an evaporator, in order to obtain information about the change in vapor content along the evaporator tube length. These relations are obtained from an experiment [4]; but analysis of the test data and their generalization are hindered due to the absence of information characterizing the ratio of vapor and fluid velocities and also the values of the fluid velocities. Measurement of these quantities in a boiling twisted flow is practically impossible. This paper is devoted to description of the mathematical model of evaporation of a twisted cryogenic fluid flow supplemented, first, by consideration of and allowance for the initial section, and, second, by calculation of the drag in the vapor gap and the calculation of the coefficient which reflects the difference of the heat transfer process in a vapor gap from molecular thermal conductivity, and also of the results of employment of this model as a source of information about the initial vapor content (at the channel inlet) and about the change in vapor content along the channel length.

An experimental investigation of boiling of a nitrogen twisted flow, as a result of which graphs of the dependence of the evaporator tube wall temperature T_w on the tube length under different conditions at the channel inlet and at different densities of the supplied heat flux q_w were obtained, is described in [5]. Thus, the cryoagent flow rate, the supplied heat flux (in [5] the value of q_w is constant along the tube length), the inlet pressure and the geometric characteristics of the tube, i.e., its length and inner diameter d_{in}, and the insert characteristics, viz., its thickness and the pitch of twisting s, are the initial parameters in the program implementing the mathematical model of the flow and heat transfer of a cryogenic fluid twisted flow. The variable initial parameters in the program are the temperature of the cryoagent at the inlet T_{inl} and the mass vapor content at the inlet x_{inl}. The aim of testing of the model as an instrument for solving the problem of evaporator optimization is determination of the values of T_{inl} and x_{inl} at which the field of the wall temperature T_w coincides along the channel length.

The originality of this paper is in the consideration of postcrisis boiling in a laminated flow with the Leidenfrost effect as one possible mode of boiling of a cryogenic fluid twisted flow. Here, the mathematical formulation of the laws governing postcrisis boiling is based on a physical model of the given regime developed earlier [2].

The mathematical model of the process of evaporation of the twisted flow of a cryogenic fluid, the results of calculation by which are compared with the results of an experimental study, involves the following possible regimes of heat transfer: convective, nucleate boiling, film-bubbling, and Leidenfrost film boiling. The physical picture of the latter regime is described in [2]. The calculation was limited by the geometric site of the transition

Kazan State Technical University. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 69, No. 5, pp. 741-747, September-October, 1996. Original article submitted February 8, 1995.

 1062-0125/96/6905-0558$15.00

(along the evaporator tube length) from Leidenfrost film boiling to a disperse regime, which was determined by a formula obtained by analysis of experimental data on heat transfer of boiling twisted flows of cryoagents [3]. If in the initial data the value of T_{inl} is smaller than the saturation temperature T_{sat}, then the tube length necessary for heat-transfer-agent heating is determined. In this section, heat transfer to a single-phase flow probably takes place, and the coefficient of heat transfer in a flow in an α_{in}-diameter channnel with a twisted belt is determined as a function of the flow mode according to [6, 7]. Heat transfer in the initial section of the considered channel is convective until the temperature difference $(T_w - T_{sat})$ reaches the value of ΔT_b that characterizes transition to nucleate boiling [8, 9]. Pressure losses in each calculated cross-section of the given region of convective heat transfer are caused by the pressure drop due to gravitation; head losses are caused by flow acceleration and pressure losses are determined by friction. The drag is found with allowance for the possible flow mode [6, 10].

In nucleate boiling the heat transfer rate is determined by the interaction of factors characterizing the heat-transfer rate due to vapor generation α_v and due to the factors of hydraulic interaction caused by forced convection α_w using the Labuntsov technique.

With the onset of a boiling crisis, we assume that a transition to the film-bubbling regime takes place, in which heat is transferred mainly by conduction and convection through a vapor film or by Leidenfrost film boiling.

As a boiling flow of a cryogenic heat carrier moves in a channel with twisting, the onset of a boiling crisis is affected by both the forced motion of the heat-transfer agent and the inertial acceleration.

The effect of the conditions of forced motion is an increase in the flow velocity relative to the wall, which leads to intense washing of vapor bubbles off the wall and to turbulization of the near-wall layer. This mechanism takes place during boiling in straight tubes. Inertial acceleration governs the formation of macrovortex structures, increases the degree of turbulence, and determines the Archimedes effect.

The change in the value of q_{cr} as function of the intensity of inertial acceleration was found by the Kutateladze technique. The inertial mass acceleration γ for calculation of the critical heat flux q_{cr} is determined with allowance for flow twisting by the formula

$$j = \frac{\pi v_{fl}}{2d_{in}} \frac{1}{(s/d_{in})^2}, \tag{1}$$

here v_{fl} is the fluid velocity in the given section of the tube in nucleate boiling.

At a certain value q_{cr}, the critical temperature drop Δt_{cr} and the critical coefficient of heat transfer α_{cr} are determined by the dimensionless relation obtained by G. N. Kruzhilin [11, 12]. At the onset of a boiling crisis, a transition is possible to either the film-bubbling regime of boiling or Leidenfrost film boiling.

The program is constructed so that after the fact of a boiling crisis, we perform calculations by the model of Leidenfrost film boiling, whose existence is indicated by the presence of large temperature differences in the experiments of [5] that greatly exceed those given in [13, 14]. The fulfillment of the relation

$$L \leq L_{lim}, \tag{2}$$

where L is the path length of vapor formed between the heated surface and the surface of the fluid film facing it, is a condition for the existence of Leidenfrost film boiling. The value of L_{lim} is calculated by the formula [15]

$$L_{lim} = \frac{3\eta v r^2 \rho_v^3 \bar{w}^3}{\left(P_{fl} - \frac{\rho_v \bar{w}^2}{2}\right) q_{ev}^2}, \tag{3}$$

where $\bar{w}$ is the velocity of the vapor escaping from under the fluid film; q_{ev} is the density of heat flux on the evaporation surface, P_{fl} is the inertial mass force of a fluid layer with thickness δ $(P_{fl} = \rho_{fl} j\delta)$, η is a coefficient allowing for the change in friction losses due to adjustment of the velocity profile in the initial section γ_1, for injection γ_2, for wave formation on the fluid film surface γ_3, and for the adjustment of the velocity profile on the heating surfaces [16].

One of the main points of the developed mathematical model of the evaporation of the twisted flow of a cryogenic fluid is the determination of velocity $\overline{w}$, which is performed assuming that the force pressing the fluid film is counterbalanced by the excess pressure of vapor under the film. The value of the latter, in the general case, is determined by friction losses and the pressure drop caused by acceleration of the vapor flow. In the model, the excess pressure Δp under the fluid film was calculated by the formula

$$\Delta p = \frac{\rho_v \overline{w}^{*2}}{2}, \tag{4}$$

where $\overline{w}^*$ is the mean velocity of the vapor escaping from under the fluid film, thus leading to the introduction of the coefficient $k < 1$, which allows for losses due to friction and the effect of the wavy character of the fluid film surface ($\overline{w} = k\overline{w}^*$).

The value of k was determined as follows: at an arbitrarily assigned value of k ($k = 1$) the level of pressure losses due to friction was found. The coefficient k uniquely determines the escape velocity $\overline{w}$ and the height of the vapor gap h under conditions of constancy of the heat transferred through the gap. For this purpose, a section with a length equal to the half-length of the channel formed by the fluid film and the wall of the vapor-generating tube was divided to n subsections, in each of which the coefficient b was found ($b = \rho w_0/\rho w_\infty$, ρw_0 is the density of the mass flow from the fluid film surface, ρw_∞ is the density of the mass flow in the vapor gap). For supercritical injections ($b \geq 0.02$ [17]), the coefficient of hydraulic resistance was calculated by the relation from [15, 18].

When $b < 0.02$, the pressure drop caused by friction and wave formation was determined in each subsection over L [15]. At the found value of Δp_{fr}, over length $L = \pi(d - 2h)/4$, the escape velocity

$$\overline{w}' = \sqrt{\left(\frac{2\,(P_{fl} - \Delta p_{fr})}{\rho_v}\right)} \tag{5}$$

and a new value of the coefficient k

$$k = \overline{w}' \Big/ \sqrt{\left(\frac{2P_{fl}}{\rho_v}\right)}. \tag{6}$$

are determined. If the difference $(k - k')$ is greater than or equal to some given value of a, we assume $k = k'$ and the calculation is repeated until the inequality $(k - k') \leq a$ is fulfilled.

The wall temperature of the vapor-generating channel T_w is found by the formula

$$T_w = T_{sat} + q_w/(M\lambda_v/h + G_1 c_p/(\pi d_{in}\,\Delta z)), \tag{7}$$

where G_1 is the flow rate of the vapor formed under the film, Δz is the length of the subsection along the axis of the pipeline, m is a coefficient reflecting the difference between the processes of heat transfer and molecular heat conduction ($m = \lambda_{ef}/\lambda$, λ_{ef} is the thermal conductivity in the gap h). For a gap h, the dynamics of the change in $\lambda_{ef} = f(b)$ is determined in a circumferential direction of motion of vapor formed in the gap and traveling length L. When $b < 0.02$, the coefficient M is determined by the formula from [19]. For $b < 0.02$, the value of λ_{ef} is determined by the formula

$$\lambda_{ef} = \frac{2\gamma_1 h}{1 + \frac{1}{\gamma_2\gamma_3}}, \tag{8}$$

where γ_3 is a coefficient allowing for the increase in heat supply to the fluid film due to the wavy character of its surface, according to P. L. Kapitsa. The coefficient γ_3 is equal to 1.21 [20].

After λ_{ef} is found for super- and subcritical injection (i.e., in each subsection over a section of length L), the mean value of λ_{ef} in the gap under the fluid film and the value of the coefficient M in the gap are determined.

The length of section l_{tr}, which is characterized by the presence of Leidenfrost film boiling, is determined by the relation from [3] given below, which was obtained by analysis of the mathematical model and the test data of [4, 5]:

$$\bar{l} = \frac{1}{2} \mathrm{Bo}^{-0.65} (s/d_{in})^{0.3}, \tag{9}$$

here $\bar{l} = l_{tr}/d_{in}$. Formula (9) was derived for nitrogen within the range of the boiling parameter Bo, whose physical meaning is the ratio of the transverse mass velocity of the vapor formed on the wall to the mass velocity of the flow in the axial direction, that corresponded to an initial section of $(0.29-3.197)\cdot 10^{-3}$ and $(s/d_{in}) = 3-8.5$.

We presented the results of calculation by a model limited to Leidenfrost film boiling.

The film-bubbling regime of boiling probably precedes Leidenfrost film boiling. The calculation program is constructed so that in the case of nonfulfillment of condition (2) one must perform the calculation by the model of film-bubbling boiling. Then the Leidenfrost film boiling is calculated.

In [9, 21-24] approaches to the calculation of heat transfer in film boiling of a cryogenic heat-transfer agent are presented that are based, for example, on the Nu number, calculated by the formulas for single-phase flow, and the Martinelli parameter. However, due to the limited information about the results of experimental studies of the considered cryogenic fluid flow, the calculation scheme for the given regime is based on its physical model with a number of assumptions.

In constructing a model of a film-bubbling regime we assume that the entire volume of vapor formed in the vapor gap between the channel wall and the twisted fluid flow moves in an axial direction and its velocity w' is equal to the axial velocity of the fluid. Then the formula for determination of the vapor gap is written in the form

$$h = \frac{2 q_w \Delta z}{w' \rho_v r}. \tag{10}$$

Here we assume that the heat supplied from the wall is spent only for evaporation.

In the first step of the calculation of temperature T_w, the coefficient of heat transfer in the vapor gap is determined by the formula

$$\alpha_{f.-b.} = \lambda_{v(T_w + T_{sat})/2}/h \tag{11}$$

and the thermophysical properties of the vapor in the gap are calculated (at temperature $(T_w + T_{sat})/2$). Then we find the coefficient $\alpha_{f.-b.}$ from the effective thermal conductivity $\lambda_{ef\,f.-b.}$, which allows for heat transfer enhancement in the vapor gap due to the difference between temperatures T_w and T_{sat} (the Mikheev formula [15]). The obtained value of T_w' is compared with its previous value T_w; if the difference exceeds the prescribed value, then we assume $T_w = T_w'$, and the iteration process is resumed until the required accuracy of calculation of the wall temperature is attained.

In real systems, due to high surface temperatures, radiation constitutes a substantial part of the total heat flow from the surface in film boiling. This effect was discussed in [9], where it was shown that for a surface temperature not exceeding 400–425 K in film boiling of oxygen, nitrogen, hydrogen, and helium, the maximum error caused by the neglect of radiation does not exceed 5%; therefore, radiative heat transfer is ignored in the model.

The flow rate of vapor formed in a section is determined with allowance for its superheating

$$\Delta G_v = \frac{q_w \pi d_{in} \Delta z}{2\,(r + 0.5 c_{p(T_w + T_{sat})/2}\,(T_w - T_{sat}))}, \tag{12}$$

then we determine the flow rates of the vapor and fluid in the section, the mass vapor content, the areas occupied by the vapor (without regard for the vapor gap) and fluid, and the vapor velocity in the flow core. The fluid velocity is calculated on the basis of equality of the change in momentum to the force impulse [1]. Losses of total pressure

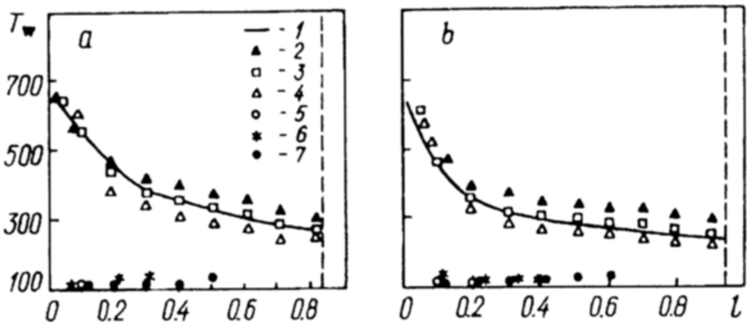

Fig. 1. Change in wall temperature along the length of a vapor-generating channel: 1) experimental results; 2, 3, 4) $T_{inl} = T_{sat}$, $x_{inl} = 0.1$ (2), 0.05 (3), 0.02 (4); 5, 6, 7) $x_{inl} = 0$, $\Delta T = 5$ K (5), 10 (6), 15 (7). Inlet conditions: a) $l_{tr} = 0.83$ m, $g = 1008.4$ kg/(m^2·sec), $q_w = 160.87 \cdot 10^3$ W/m^2, $T_{inl} = 91$ K, $p_{inl} = 3.83 \cdot 10^5$ Pa; b) $l_{tr} = 0.95$ m, $g = 1005.8$ kg/(m^2·sec), $q_w = 196.07 \cdot 10^3$ W/m^2; $T_{inl} = 92$ K, $p_{inl} = 4.03 \cdot 10^5$ Pa. T_w, K; l, m.

are calculated as losses caused by viscous friction, the effect of the mass force, and the change in momentum in the flow [1].

The graphs show experimental and calculated regularities of the change in the wall temperature along channel length l. Calculations were performed at different values of mass vapor content and temperature of the medium at the inlet.

Figure 1 illustrates the dynamics of the change in temperature T_w along the tube length for three values of vapor content at the inlet at $T_{inl} = T_{sat}$ and three values of flow undercooling at the inlet at $x_{inl} = 0$.

The level of the wall temperature decreases with the vapor content at the inlet. As fluid undercooling at the channel inlet increases, the length of the zone of convective heat transfer increases with a decrease in the level of the wall temperature. These regimes did not take place in the experiments, therefore, calculation by the model with undercoolings over the entire tube length was not performed.

The results of calculations and their comparison with the experimental data allowed one to determine the mass vapor content at the channel inlet (consequently, the dynamics of its change along the length l), which was observed under the corresponding experimental conditions of boiling of a twisted flow of nitrogen. Thus, for the conditions illustrated by Fig. 1, coincidence of calculations with the experiment in both cases was found at $x_{inl} = 0.05$. Here, in the first case the maximum difference does not exceed 7%; in the second case, 10%.

Thus, comparison of the results of the calculation of thermal processes by the mathematical model of the evaporating twisted flow of cryogenic fluid with the results of the experimental study of the heat transfer rate of a nitrogen flow in a tube with a twisted belt showed their satisfactory agreement. This justifies use of the model and the programs implementing it for optimal design of evaporators of cryogenic fluids.

NOTATION

g, density of mass flow at channel inlet; q_w, density of supplied heat flux; λ, thermal conductivity of vapor in the gap; c_p, specific heat of vapor in the gap; r, latent heat of vapor generation; ν, coefficient of fluid kinematic viscosity; ρ_v, vapor density; ρ_{fl}, fluid density; G_l, vapor flow rate; ΔT, fluid undercooling to the saturation temperature. Indices: w, wall; in, inner; inl, inlet; sat, saturation; b, bubble; v, vapor; c, convection; fr, friction; fl, fluid; cr, critical; lim, limiting; ef, effective; tr, transition; f.-b., film-bubbling; ev, evaporation.

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