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HEAT TRANSFER DURING SUBCOOLED BOILING IN TUBES (A REVIEW)

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This article provides a review of the correlations and models for determining the intensity of heat transfer during subcooled boiling in pipes. As a rule, correlations are based on dimensionless similarity numbers, while heat exchange models with subcooled boiling use the principle of superposition of the components of heat transfer during forced convection and developed nucleate boiling. Various authors propose different approaches to the implementation of the principle of superposition. This article presents an analysis of the advantages and disadvantages of the correlations and models. These advantages and disadvantages were determined both by analyzing the physical laws of subcooled boiling and by comparing the results that were obtained by the authors of this article by means of various models of subcooled boiling with the experimental data obtained during the study of heat transfer during the subcooled boiling of ammonia in a cylindrical heated tube. The tube diameter d was 6.9 mm, length L was 150 mm, inlet subcooling was ~5 K, saturation temperature range was 61...65 °C, mass flow rate was 7.5 g/s, and heat flux density range was 5...18 W/cm^2. As a result of the review and comparison with the experimental data, it was determined that the existing correlations and models describe the subcooled boiling of ammonia with insufficient accuracy, especially in the area of the combined effect of forced convection and nucleate boiling. Therefore, it is necessary either to refine the existing correlations and models, or develop new models for a more precise description of the subcooled boiling heat transfer of ammonia in heated tubes in the parameter range specified above.

Key words: *subcooled boiling, nucleate boiling, forced convection, models of subcooled boiling, principle of superposition, ammonia.*

Introduction

The use of heat transfer with developed nucleate boiling allows obtaining high heat transfer coefficients and, accordingly, minimizing the size and mass of heat transfer equipment. Often, the heat transfer device receives a subcooled liquid, which is why, in practice, it is necessary to determine the heat transfer intensity in the transition region between the heat transfer during the motion of a single-phase liquid and heat transfer during developed boiling. In this region, the temperature of the heated surface may exceed the saturation temperature, which will lead to boiling on the surface, despite the fact that the fluid in the core of the flow will have a temperature below the saturation temperature. Heat transfer during boiling in a liquid subcooled to the saturated state is not fully studied, although the heat transfer coefficient during subcooled liquid boiling may increase several times (up to 20 times under certain conditions) compared with that in a single-phase liquid.

Purpose of the Work and Formulation of the Research Problem

The task is to perform a review of the known correlations and models that are used in determining the heat transfer under subcooled boiling, determine their advantages and disadvantages, compare them with the experimental data obtained by the authors and, based on the review, identify ways of refining the existing models or developing new ones for the subcooled boiling heat transfer in heated channels.

Correlations and Models of Subcooled Boiling Heat Transfer

Correlations and models of subcooled boiling heat transfer can be divided into two groups: group 1 where the correlations are based on simple dimensionless ratios, and group 2 where the heat transfer components during convection and boiling are calculated separately and then added according to certain methods.

The correlations of the 1st group can be said to include the correlation of Moles & Shaw [1]

$$\frac{h_{TP}}{h_L} = 78.5\left(\mathrm{Bo}\cdot\mathrm{Ja}^{-0.746}\left(\frac{\rho_L}{\rho_v}\right)^{0.045}\mathrm{Pr}_L^{0.69}\right)^{0.67},$$

where h_{TP} is the subcooled boiling heat transfer coefficient, W/(m^2·K); h_L is the forced convection heat transfer coefficient, W/(m^2·K); $\mathrm{Bo} = \frac{q}{G_f r_{fg}}$ is the boiling number; $\mathrm{Ja} = \frac{c_{pL} \Delta T_{sub}}{r_{fg}}$ is the Jacob number; ρ_L is the liquid phase density, kg/m^3; ρv is the vapor phase density, kg/m^3; Pr_L is the Prandtl number of the liquid phase.

The heat flux density is determined by the temperature pressure $(T_w - T_{sat})$ and, accordingly, will be $q = h_{TP}(T_w - T_{sat})$. Here, T_w is the wall temperature, K; T_{sat} is the saturation temperature, K.

There are other correlations for calculating the subcooled boiling heat transfer coefficient, which have a similar structure. Thus, in work [2], one of such correlations was proposed in the following form:

$$\frac{h_{TP}}{h_L} = \exp(14.542)\mathrm{Bo}^{0.729} \cdot \mathrm{Ja}^{-0.354} \left(\frac{\rho_L}{\rho_v} \right)^{1.811} \mathrm{Pr}_L^{7.032}$$

The main disadvantages of the correlations of the 1st group are their failure to be used in the entire area of subcooled boiling, because with decreasing subcooling the Jacob number $\mathrm{Ja} \to 0$. As a result, $\frac{h_{TP}}{h_L} \to \infty$. In addition, the right-hand side of the correlations does not tend to unity at the point of transition from the forced convection heat transfer to subcooled boiling heat transfer.

The design algorithms of the 2nd group can be said to conditionally include the model described in [3].

To take into account the influence of the forced motion on the intensity of heat transfer, the author considers three zones whose boundaries are determined by the ratio between the developed nucleate boiling heat transfer coefficient and the forced fluid convection heat transfer coefficient. This ratio is $A = {h_q}/{h_L}$, where h_q is the developed nucleate boiling heat transfer coefficient, W/(m^2·K).

In zone 1, ${h_q}/{h_L} < 0.5$, $h_{TP} = h_L$, boiling does not affect heat transfer. The heat transfer intensity is completely determined by the convection of the liquid.

In zone 2, ${h_q}/{h_L} > 2$, $h_{TP} = h_q$, the heat transfer intensity is fully determined by boiling in a large volume; forced convection does not affect heat transfer.

Zone 3, $0.5 \le {h_q}/{h_L} \le 2$, is the boiling area with the mutual influence of convective heat transfer and developed boiling heat transfer; the expression for the heat transfer coefficient is described by the interpolation formula

$$h_{TP} = h_L \frac{4h_L + h_q}{5h_L - h_q}$$

Despite the fact that this algorithm is proposed for calculating the intensity of the saturated liquid boiling heat transfer, it can also be used to analyze the subcooled boiling heat transfer intensity. It is proposed to determine the subcooled boiling heat flux density by the formula $q = h_{TP}(T_w - T_{sat})$.

Other methods for determining the intensity of subcooled liquid boiling heat transfer are based on the superposition of heat transfer components. Such an approach was proposed for determining the heat transfer coefficient during developed nucleate boiling in [4] and then modified for the case of subcooled boiling. One of such methods is considered in [5] where heat transfer is observed during boiling in a subcooled liquid, in pipes, and in annular channels. The advantage of this method is that the zones of various heat transfer mechanisms are not specifically identified in advance. The heat transfer coefficient is determined by the method of superposition of two heat transfer components: the convective component and the boiling component. In this case, the subcooled boiling heat transfer coefficient will be [6]

$$h_{TP}^2 = (Fh_L)^2 + (Sh_q)^2.$$

Thus, the heat transfer coefficient is determined by the ratio between the various mechanisms of heat transfer. For example, the convective component will prevail over the boiling component Fh_L if it is much larger than Sh_q, and vice versa.

The authors of this work point out that in the case of subcooled boiling, the temperature driving forces for the convective and boiling components are different. Therefore, the equation for the heat flux density is represented as

$$q_{pb} = \sqrt{(Fh_L \Delta T_b)^2 + (Sh_q \Delta T_s)^2}, \quad (1)$$

where q_{pb} is the subcooled boiling heat flux density, W/m^2; $\Delta T_b = T_w - T_L$ is the difference between the wall and subcooled liquid temperature, K; $\Delta T_s = T_w - T_{sat}$ is the difference between the wall temperature and the saturation temperature, K.

Because the heat transfer coefficient for nucleate boiling depends on the heat flux density, the authors represent equation (1) as $q_{pb} = \sqrt{(Fh_L \Delta T_b)^2 + \left(SA_p q_{pb}^{2/3} \Delta T_s\right)^2}$. The equation is then written in the form of the cubic equation $q_*^3 - Cq_*^2 - 1 = 0$ with respect to the dimensionless parameter $q_*^3 = \left(\frac{q_{pb}}{q_L}\right)^2$. Here, $q_L = Fh_L \Delta T_b$ and $C = \left(\frac{A_p S \Delta T_s}{Fh_L \Delta T_b}\right)^2 q_L^{4/3}$.

Thus, if the channel geometry, mass flow rate, wall temperature T_w, temperature T_L, and saturation temperature T_{sat} are known, then, when solving the cubic equation, we find q_* and then determine the subcooled boiling heat transfer coefficient $h_{TP} = Fh_L q_*^{3/2}$.

If the heat flow density q_{pb} is given, then, by solving equation (1) with respect to the temperature driving force between the wall and the liquid volume, $\Delta T_b = T_w - T_L$, we obtain

$$\Delta T_b = \frac{T_{sat} - T_b}{1 + A_{bp}^2}\left[1 + \sqrt{1 + \left(1 + A_{bp}^2\right)\left(A_{qp}^2 - 1\right)}\right],$$

where $A_{bp} = \frac{Fh_L}{Sh_q}$ and $A_{qp} = \frac{q_{pb}}{Sh_q (T_{sat} - T_L)}$.

Finally, the subcooled boiling heat transfer coefficient will be $h_{TP} = \frac{q_{TP}}{\Delta T_b}$.

Work [7] also uses the superposition method, but in a simplified form. The essence of the method is illustrated in Fig. 1.

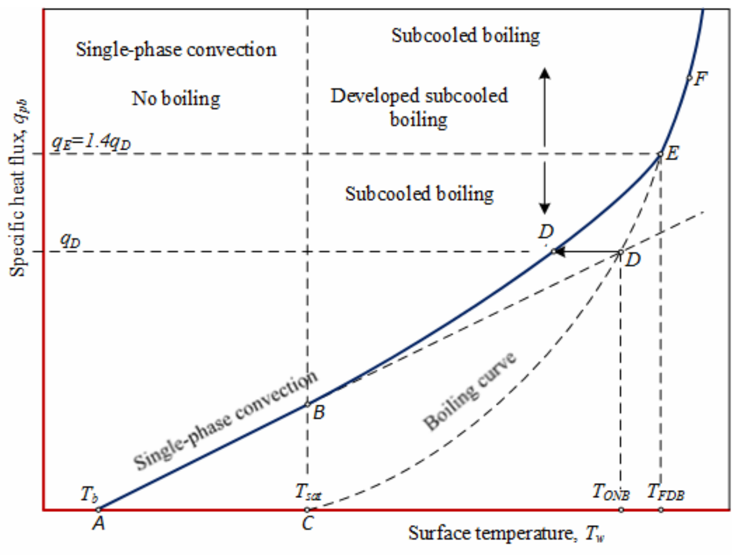

Fig. 1. Dependence q_{pb}=f(T_w) in accordance with model [7]

With increasing heat flux, the wall temperature changes along the *ABD* line, which corresponds to convective heat transfer. In this case, the specific heat flux is $q_{SPL} = h_L (T_{w,SPL} - T_L)$. When the wall temperature reaches the boiling point T_{ONB} (point *D* in Fig. 2), vapor bubbles start forming on the wall, the heat transfer coefficient increases, with the wall temperature decreasing spasmodically (point *D*').

With further increase in heat flow, the temperature change is characterized by the *D'EF* curve. Point *E* corresponds to the start of fully developed nucleate boiling. In this case, the temperature driving force, depending on the heat flux density, can be represented by the generalized dependence $T_{w,SCB} - T_{sat} = \psi q_{SCB}^n$.

In accordance with the proposed model, the initial subcooled boiling point (point D in Fig. 1) is determined by the intersection of the ABD and CDE curves, that is, it follows from the condition $T_{w,SPL} = T_{w,SCB}$ that $T_L + \frac{q_{pb}}{h_L} = T_{sat} + \psi q_{pb}^n$.

As a result, the subcooling at the initial boiling point

$$(T_{sat} - T_L)_{ONB} = \Delta T_{ONB} = \frac{q_{pb}}{h_L} - \psi q_{pb}^n . \quad (2)$$

The position of point E, corresponding to fully developed nucleate boiling, is defined as $q_E = 1.4 q_D$ [8]. Or, by analogy with equation (2),

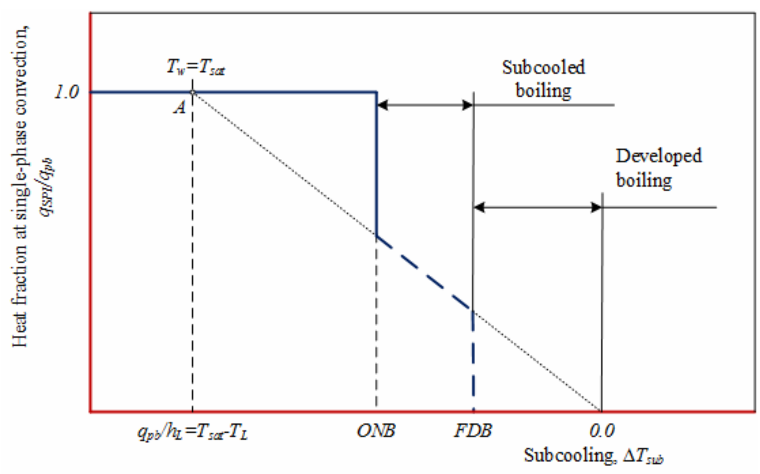

Fig. 2. Dependence of heat fraction at single-phase convection on subcoolings [7]

$$\Delta T_{FDB} = (T_{sat} - T_L)_{FDB} = \left(\frac{q_{pb}}{1.4 h_L}\right) - \psi \left(\frac{q_{pb}}{1.4 h_L}\right)^n .$$

The calculated values of the liquid subcooling up to the saturation temperature $\Delta T_{sub} = (T_{sat} - T_L)$ are used to determine the boundaries of various heat exchange mechanisms. For this, it is proposed to use the dependence of the heat flux fraction during convection on the total heat flux during subcooled boiling (see Fig. 2).

In Fig. 2, the following zones are identified (the graph shows a leftward increase in subcooling):

– if the subcooling is more than $\Delta T_{sub} > \Delta T_{ONB}$, the heat transfer is characterized by single-phase convection, even if $T_w > T_{sat}$;

– if the subcooling $\Delta T_{FDB} \le \Delta T_{sub} < \Delta T_{ONB}$, the convective heat flux and heat flux during boiling are determined taking into account the diagram in Fig. 3 (dashed line between the *ONB* and *FDB* points);

– if the subcooling $\Delta T_{sub} \le \Delta T_{FDB}$, then $q_{SPL} = 0.0$ and the heat transfer intensity is determined only by boiling.

A similar method (superposition method) is proposed in [9], according to which $q_{pb} = q_{SPL} + q_{SCB}$. The heat flow at single-phase convection is determined as $q_{SPL} = h_L (T_w - T_L)$. Consequently, the heat flux at boiling $q_{SCB} = q_{pb} - q_{SPL}$. In this article, the heat transfer coefficient or the temperature driving force are determined by the value q_{SCB}, using one of the correlations for heat exchange during nucleate boiling. Thus, this paper uses the ratio

$$\frac{c_{pL}(T_w - T_{sat})}{r_{fg}} = C_{sf} \left[\frac{q_{SCB}}{\mu_L r_{fg}} \sqrt{\frac{\sigma}{g(\rho_L - \rho_v)}} \right]^{0.33} \left(\frac{c_{pL}\mu_L}{k_L} \right)^{1.7} .$$

The expression for calculating the intensity of heat transfer during boiling in the general case can be represented as $q_{SCB} = \left(\frac{T_w - T_{sat}}{\psi}\right)^{\frac{1}{n}}$. From this relation and the relation $q_{SPL} = h_L (T_w - T_b)$, we find the wall temperature T_w, solving the transcendental equation $q_{pb} = h_L (T_w - T_b) + \left(\frac{T_w - T_{sat}}{\psi}\right)^{\frac{1}{n}}$ with respect to T_w. By setting the temperature driving force, for example, $\Delta T_{sat} = T_w - T_{sat}$, we determine the subcooled boiling heat transfer coefficient $h_{TP} = \dfrac{q_{SCB}}{\Delta T_{sat}}$.

The following model is a continuation of model [9] and is described in [10]. The essence of the method is illustrated in Fig. 3. In accordance with the model, the heat flow during subcooled boiling

$$q_{pb} = q_{SPL}\left[1 + \left\{\frac{q_{SCB}}{q_{SPL}}\left(1 - \frac{q_{C''}}{q_{SCB}}\right)\right\}^2\right]^{\frac{1}{2}}$$

The heat flux $q_{C''}$ corresponding to point C'' is determined from the equation of the boiling curve $T_{w,ONB} - T_{sat} = \psi q_{C''}^n$ with the wall temperature equal to the initial boiling point T_{ONB}. To determine the initial boiling point, the paper proposes the following empirical relations:

$$\Delta T_{ONB} = (T_w - T_{sat})_{ONB} = \left(\frac{8\sigma T_{sat}}{r_{fg} k_L \rho_v}\right)^{1/2} \cdot \Pr_L q_{ONB}^{\frac{1}{2}}.$$

In this case, the heat flow $q_{ONB} = h_L[\Delta T_{ONB} + \Delta T_{sub}]$.

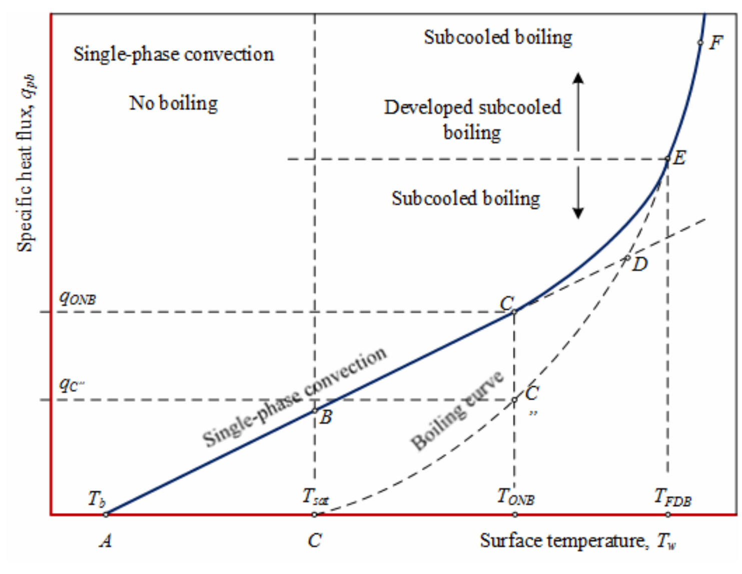

Fig. 3. Dependence $q_{pb}=f(T_w)$ in accordance with the model [10]

In [11], it is proposed to approximate the curve between points C and E (see Fig. 3) by the power dependence $q_{pb} = a + b(T_w - T_{sat})^m$. The constants a, b, m are defined as

$$b = \frac{q_E - q_c}{(\Delta T_{sat,E})^m - (\Delta T_{sat,C})^m},$$

$$a = q_c - b(\Delta T_{sat,C})^m, \quad m = n + p q_{pb},$$

$$p = \frac{1/0.3 - 1}{q_E - q_C} \text{ and } n = 1 - p q_C.$$

The interpolation proposed provides for a smooth transition from one heat exchange mode to another. In addition, the curve slope changes with increasing wall temperature, which corresponds to the physical picture of an increase in the intensity of heat exchange with an increase in the number of evaporation centers.

For the calculation of heat fluxes at points C and E, the author proposes his own correlations. For example, the heat flux during developed nucleate boiling (point E) is determined by the formula $q_E = \left[1058\left(m \cdot r_{fg}\right)^{-0.7} F_{fl} h_L \Delta T_{sat,E}\right]^{\frac{1}{0.3}}$.

The heat flux at the initial boiling point (point C) is determined using relations [12, 13]

$$\Delta T_{sat,C} = \frac{4\sigma T_{sat}(v_v - v_L)h_L}{k_L r_{fg}}\left[1 + \sqrt{1 + \frac{k_L r_{fg} \Delta T_{sub}}{2\sigma T_{sat}(v_v - v_L)h_L}}\right] \quad q_C = \frac{k_L r_{fg}}{8\sigma T_{sat}(v_v - v_L)}[\Delta T_{sat,C}]^2.$$

Model Analysis

In this analysis, we will compare different models of subcooled liquid boiling heat transfer with experimental data (see Fig. 4). The models based on the dimensionless correlations of Moles & Shaw et al. are not considered due to their disadvantages presented above.

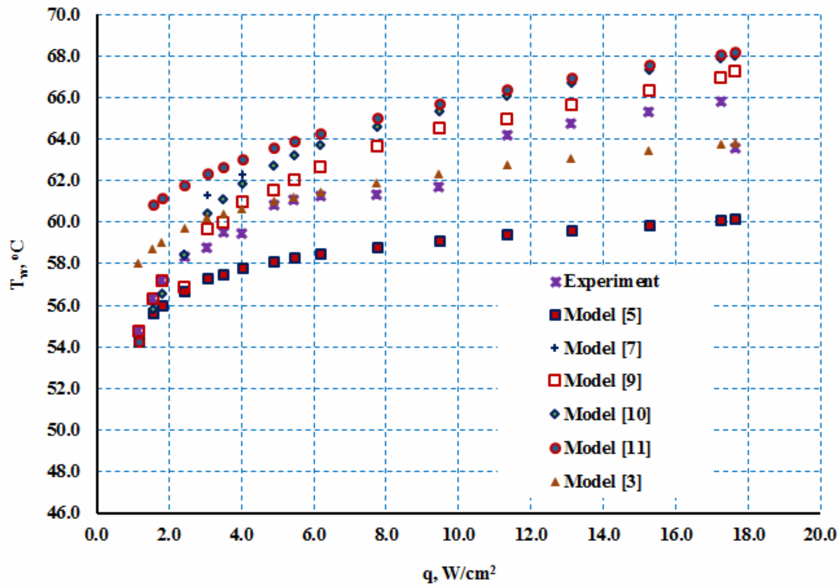

Fig. 4. Comparison of the experimental results with the results obtained using different models of heat transfer during subcooled boiling

The experimental data were obtained by the authors of this article in studying the flow of ammonia in a heated cylindrical channel with the diameter d=6.9 mm and length L=150 mm. In the experiments, the subcooled boiling at the channel inlet was ~5 K, the saturation temperature T_{sat} was 61–65 °C, the mass flow rate m was 7.5 g/s, and the heat flux density q was 5–18 W/cm^2.

Conclusions

1. The main disadvantages of the correlations based on simple dimensionless ratios are their failure to be used in the whole area of subcooled boiling, because with decreasing subcooling $\mathrm{Ja} \to 0$. As a result $\frac{h_{TP}}{h_L} \to \infty$. In addition, the right-hand side of the correlations does not tend to unity at the point of transition from the forced convection heat transfer to subcooled boiling heat transfer.

2. It is not indicated to what temperature driving force the heat transfer coefficient refers.

3. Among the models based on superposition, the Rohsenow model gives the best agreement with the experimental results. This model is also the simplest one compared to others.

4. The models of Labuntsov and Liu & Winterton give underestimated wall temperature values. In addition, the disadvantage of the Labuntsov model is that it requires the specification of heat transfer boundaries, which must be chosen and justified for each liquid and the range of flow parameters.

5. The disadvantage of the Kandlikar, Bergles & Rohsenow models is that for their realization, it is necessary to know the wall temperature at the initial boiling point. To search for this temperature, the model developers use empirical relationships, which also need to be justified.

6. The comparison with the experimental data showed that all the models considered describe the heat exchange with the subcooled boiling of ammonia in cylindrical pipes in the above parameter range with a large error. Therefore, the models must either be refined, or new ones must be developed.

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Теплопередача при недогрітому кипінні в трубах (огляд)

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В статті наведено огляд кореляцій та моделей для визначення інтенсивності теплообміну під час недогрітого кипіння теплоносія у трубах. Кореляції, як правило, ґрунтуються на безрозмірних числах подібності, в той час як в моделях недогрітого кипіння використовується принцип суперпозиції складових теплообміну під час вимушеної конвекції та розвиненого пузиркового кипіння. Різними авторами запропоновано різні підходи до реалізації принципу суперпозиції. В статті виконано аналіз переваг та недоліків кореляцій та моделей. Переваги та недоліки визначалися як шляхом аналізу фізичних закономірностей недогрітого кипіння, так і порівнянням результатів, одержаних авторами статті за допомогою різноманітних моделей з експериментальними даними під час дослідження недогрітого кипіння аміаку в циліндричній трубі, що обігрівалась. Діаметр експериментальної труби складав d=6,9 мм, довжина L=150 мм, недогрів на вході дорівнював ~5 K, температура насичення знаходилася в діапазоні 61–65 °C, масова витрата складала 7,5 г/с, густина теплового потоку знаходилася в діапазоні 5–18 Вт/см^2. В результаті огляду та порівняння з експериментальними даними визначено, що існуючі кореляції та моделі описують недогріте кипіння аміаку з недостатньою точністю, особливо у разі спільного впливу вимушеної конвекції та пузиркового кипіння. Тому необхідно або уточнювати існуючі кореляції і моделі, або розробляти нові моделі для більш точного опису теплообміну під час недогрітого кипіння аміаку в трубах, що обігріваються, у вказаному вище діапазоні параметрів.

Ключові слова: *недогріте кипіння, пузиркове кипіння, вимушена конвекція, моделі недогрітого кипіння, принцип суперпозиції, аміак.*

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STRESSED STATE OF A HOLLOW CYLINDER WITH A SYSTEM OF CRACKS UNDER LONGITUDINAL SHEAR HARMONIC OSCILLATIONS

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This paper solves the problem of determining the stress state near cracks in an infinite hollow cylinder of arbitrary cross section during longitudinal shear oscillations. We propose an approach that allows us to separately satisfy conditions both on the cracks and boundaries of a cylinder. The problem reduces to the equations of motion in a flat domain with the defects bounded by arbitrary smooth closed curves under anti-plane deformation conditions. The solution scheme is based on the use of discontinuous solutions to the equations of motion of an elastic medium with displacement jumps on the surfaces of defects. Displacements in a cylinder with defects are represented both as a sum of discontinuous solutions constructed for each defect and an unknown specific function ensuring that the conditions of a harmonic load on the body boundaries are met. This function is sought as a linear combination of linearly independent solutions to the equations of the theory of elasticity in the frequency domain with unknown coefficients. The constructed representation makes it possible to separately satisfy the boundary conditions on the surfaces of defects, which results in a set of systems of integral equations that differ only in their right-hand sides and do not depend on the body boundary shape. The resulting systems of integral equations can be solved by the method of mechanical quadratures. After that, the conditions on the boundaries of the cylindrical body are satisfied, from which the unknown coefficients of the introduced specific function are determined by a collocation method. Using the approach proposed, the stress intensity factors in the vicinity of defects were calculated. With the help of those calculations, we investigated the effect of the frequency and location of the defects on the stress intensity coefficient values.

Keywords: *hollow cylinder, harmonic oscillations, stress intensity factors, system of cracks.*

Introduction

Investigation of the stress state of the bounded bodies with cracks is relevant both for determining the conditions for the destruction of the bodies by estimating the coefficients of the intensity of dynamic stresses in the vicinity of cracks and diagnosing such defects based on the information on their effect on resonance frequencies. The results obtained in this direction related mainly to both unbounded and semi-bounded defective bodies [1-4]. A much smaller number of situations have been considered for the cases where the bodies occupy limited areas. This is due to the fact that with the application of the method of boundary integral equations, the original problems are reduced to the interrelated systems of integral equations given on both the defect surface and boundary of the body [5–7], which significantly complicates the numerical implementation, especially in the