1. INTRODUCTION

Until recently, condensation of superheated steam on jets of strongly subcooled liquid has hardly been studied at all. But now, projects are being developed that require recommendations for constructors. In particular, it is the project of a direct contact reheater of high pressure feed water.

It has been proposed to use it as coolant superheated steam, taken directly after the steam generator before its inflow into the turbine. This solution solves the problem of the working medium reverse flow in the turbine channel in the case of a sudden...
shutdown of the turbine. The possibility of such a situation was the main reason for not using direct contact heat exchangers for high-pressure reheaters.

Namely this problem was the reason for the appearance of this work. It can be assumed that the presence of practical necessity will lead in the future to an active study of this problem, which is complicated enough.

The main disadvantage of the works devoted to the description of the condensation process on jets is the use of the characteristics of turbulence of flow in the nozzle without account for their change as a result of the mechanical interaction of liquid and vapor on the phase boundary. In the studies of Kutateladze (1949, 1963) and Celata et al. (1989) this boundary is assumed to be smooth, which allows one to ignore the effect of surface tension.

On the other hand, empirical correlations for the coefficient of heat transfer during condensation on jets show a significant effect of surface tension in terms of the Weber number. This problem was addressed in our previous paper (Gotovsky et al., 2012).

In order to consider the problem of taking into account the surface tension effect, the approach was used proposed in the work of Levich (1962). Its essence is as follows. In the neighborhood of the phase boundary, turbulent characteristics of liquid

\[ Y = 1 - 0.1\left(\frac{\rho'}{\rho''} - 1\right)^{0.4} (1 - \chi)^{0.4} \]

### NOMENCLATURE

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<thead>
<tr>
<th>Symbol</th>
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<tbody>
<tr>
<td>(a)</td>
<td>thermal diffusivity, (m^2/s)</td>
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<td>(c_p)</td>
<td>specific heat at constant pressure, (J/kg\cdot K)</td>
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<td>(h)</td>
<td>heat transfer coefficient, (W/(m^2\cdot K)) or (kW/(m^2\cdot K))</td>
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<td>(j_w)</td>
<td>wall mass flow density, (kg/m^2\cdot s)</td>
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<td>(k)</td>
<td>wave vector, (m^{-1})</td>
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<td>(Nu)</td>
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<td>(P)</td>
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<td>(q)</td>
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<td>(U)</td>
<td>flow velocity, (m/s)</td>
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<td>(X)</td>
<td>steam mass quality</td>
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### Greek symbols

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<td>(\lambda)</td>
<td>thermal conductivity, (W/(m\cdot K)) or (kW/(m\cdot K))</td>
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<td>(\mu)</td>
<td>dynamic viscosity, (Pa\cdot s)</td>
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<tr>
<td>(\rho', \rho'')</td>
<td>water and steam densities under saturation conditions, (kg/m^3)</td>
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<tr>
<td>(\sigma)</td>
<td>surface tension coefficient, (N/m)</td>
</tr>
<tr>
<td>(\tau)</td>
<td>oscillation period, (s)</td>
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<tr>
<td>(\nu)</td>
<td>molecular kinematic viscosity, (m^2/s)</td>
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### Subscripts

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<tr>
<td>(f)</td>
<td>subcooled boiling</td>
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<tr>
<td>(l)</td>
<td>liquid</td>
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<td>(p)</td>
<td>pool boiling</td>
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flow are changing. The surface tension forces that prevent deformation of the surface have inhibitory effects on the turbulent motion. In contrast to the solid surface the speed on a free surface does not become zero, but should practically be constant in depth as shear stress is close to zero. If we assume that the damping effect on the movement of the surface elements associated with the surface tension action, provides some stability to the surface, then in a certain range of the thickness of order \( b \), the movement will have a special character. Such an approach made it possible to explain the structure of empirical formulas for heat transfer during condensation of saturated steam on jets.

2. SPECIFIC FEATURES OF OUR APPROACH TO SUPERHEATED STEAM CONDENSATION ON SUBCOOLED LIQUID JETS

For condensation of superheated steam on jets of subcooled liquid in the monograph of Kutateladze (1949) it was proposed to use solutions for saturated steam, replacing the heat of the phase transition by the difference in the enthalpies of superheated steam and of saturated liquid. However, for intense processes one cannot ignore the effects of the interaction of vapor and liquid, which, in particular, are expressed in the form of accounting for the effect of surface tension and of interface unsteadiness. These aspects of the process are mainly considered in this work. It is important to note the following. In some studies it is indicated that the solution obtained there for the problem of turbulent jets accounts for the effect of surface tension. As an example, we can point to a very interesting, in general, monograph of Kholpanov and Shkadov (1990). But accounting for surface tension in this monograph is fundamentally different from the approach used by Gotovsky et al. (2012). They considered the macroscopic curvature of a jet to be connected with its expansion (or contraction). In the above dependences it comes to nonstationary motion of the boundaries, and they can be taken into account when solving the problem only in terms of the changes occurring in the characteristics of turbulence. This treatment is significantly more challenging than that utilized in Kholpanov and Shkadov (1990). Therefore, a conclusion was drawn about the very weak influence of the Weber number, which is at variance with the majority of empirical formulas.

Very interesting information on the interaction of gas flow with the surface of a liquid film was presented in Terekhov et al. (2000). It contains the results of experimental and theoretical studies of gas desorption from the surface of a liquid film. Calculations were performed with the use of the limiting relative laws of Kutateladze–Leontiev. Comparison of experimental curves for mass transfer with calculations on the assumption that the turbulent boundary layer is formed in the liquid and gas on a smooth flat interface indicates that the actual mass flow rates exceed the calculated values by more than several orders of magnitude. The authors believe that such an intense mass transfer is primarily due to the wave nature of the flow and to the
interaction of flows that generates vortex flow near the phase boundary. It should be noted that such an explanation is in good agreement with the above-mentioned model proposed in Levich (1962) (Fig. 1).

During condensation of pure saturated steam on the interface with the liquid (in the form of jets or a film), the temperature is equal to the saturation temperature. What does determine the value of the steam flux to the surface of condensation? This flux is determined by the thermal resistance, which limits the ability to remove heat from the surface into the liquid phase. What does then cause steam movement to the surface? Any force except the pressure gradient is absent here. Quantitative estimations show that for reasonable values of pressure and heat flux the pressure gradient is very small and can be neglected. But we must also ensure the reduction of the steam temperature to $T_s$, at which condensation will occur. Now let us consider as an example the equation of convective heat transfer in a clean steam, the thermal conductivity at atmospheric pressure, and superheated steam at $100^\circ\text{C}$:

$$\frac{dT}{dy} = a \frac{d^2T}{dy^2}. \quad (1)$$

FIG. 1: Comparison of calculated and experimental data adapted from Terekhov et al. (2000). Calculation by the Kutateladze–Leontiev method: 1) laminar film, 2) turbulent film; O, Δ, □) experimental results; $Re_x$, $Re$ number along the duct; $Re_s$, $Re$ number of the film.
Let \( dT/dy = Z \). Then we have \( Z' = vZ/a \), where \( Z = \exp (my) \), \( m = v/a \).

Since the velocity \( v \) is negative, we obtain \( m = -3400 \).

If we restrict the molecular thermal conductivity of steam, we obtain for \( y \) close to zero

\[
q = \lambda_v dT/dy = 0.026 \times 340,000 = 8840 \text{ W/m}^2
\]

— very small value.

But in fact, the solution obtained is not suitable for practical use. Indeed, the numerical value of the steam temperature gradient near the boundary phase is 340,000\(^\circ\)C/m.

If we assume that all the superheat is concentrated in the linear region near the phase boundary, we find that the thickness of this region is about 0.3 mm, which of course is unrealistic, taking into account the significant fluctuations on the phase boundary.

The only question is how to describe quantitatively the real picture of heat transfer at the phase boundary.

Experimental data for such conditions are practically absent. Accordingly, consideration will be conducted in two ways.

Firstly, it is an idea to try to compare the process of condensation of superheated steam on a subcooled jet with some sufficiently identical processes, which were better studied experimentally. As such processes one can use subcooled film boiling and post-CHF thermal conditions close to inverted annular flows. The quantity of experimental data for these problems is sufficient, because there are a number of practical issues, where these data are needed. It is suffice to mention quenching, reflow of reactor core, temperature regime in post-CHF zone, etc.

Secondly, it is possible to continue the analysis which was outlined in Gotovsky et al. (2012) for determining the order of the quantitative characteristics of a wavy liquid surface.

For the description of considered processes, we have the following picture illustrated in Fig. 2.

**FIG. 2:** Film boiling of subcooled liquid
The steam moves with some velocity along a solid surface, which has a very high temperature. On the phase boundary, capillary waves of oscillating type are being developed. Traditional formulas of heat transfer for film boiling are not valid for such a system. The scheme of heat transfer in these conditions has been discussed in the papers of Nigmatulin et al. (1994) and Kuzma-Kichta et al. (1995), where experimental data were obtained too. For relatively low heat fluxes the thermal resistance was determined in terms of the average film vapor thickness. In this simplest form, the wave nature of the interface was accounted for by calculating the thermal resistance for varying thickness of the film along the single wavelength. But calculations carried out in Gotovsky (2000) showed that such an approach for strong subcooling and a large thermal load does not lead to success.

More real is another scheme of description which is reduced to evaporation of liquid on the crests of waves and to condensation on their troughs. The main feature of this mechanism is some convergence with the mechanism of nucleate boiling for strongly subcooled liquid. Without going into further details of the analysis carried out in the papers of Nigmatulin et al. (1994) and Kuzma-Kichta et al. (1995), we may proceed directly to the results of experimental investigation of heat transfer in the above-mentioned similar processes. For example, convenient data for use in the present analysis may be obtained from the study of the coefficients of heat transfer in quenching (Bamberger and Prinz, 1986), which are given in Fig. 3. However, it is important to note at once that the results of this study refer, for obvious reasons, only to small pressure, but they are valuable from the point of view of the process analysis for a wide range of pressures.

From the results given by Bamberger and Prinz (1986) we can conclude that the minimum value of the coefficient of heat transfer from steam to the surface of the

FIG. 3: Test data of heat transfer in quenching adapted from Bamberger and Prinz (1986)
phase transition to the atmospheric pressure can be taken as 1500–2000 W/m²·K. For these conditions, the solution of Eq. (1) has no sense. The temperature distribution in this steam can be roughly represented as follows: in the boundary layer, corresponding to the value of the heat transfer temperature, it varies from $T_0$ to $T_S$, and then it remains practically constant.

Then the picture of the process is as follows. In a thin layer comprising a phase boundary, the temperature is $T_S$. Depending on the temperature of the jet and the internal heat transfer coefficient some part of the heat supplied to the surface is absorbed by the jet. The remaining portion of the heat goes to evaporating the outer layer of the jet. It can be expected that this part may be significant only at low subcooling of the jet. So, we have a process scheme in which evaporation and condensation occur parallelly. It should be noted that the above heat transfer coefficient values correspond to the atmospheric pressure. This leads to the need to assess the extent to which you can use the results obtained for the atmospheric pressure at least for conservative assessment of the situation at high pressures. With high probability we can assume that with increasing pressure in the proposed scheme, the heat transfer coefficients have to grow. Within the framework of conservative estimates, the effect of pressure and the presence of a finite rate of flow should be taken into account. As is known, the boiling heat transfer coefficient increases with the level of pressure very quickly. But it concerns only nucleate boiling, which is absent in this case. For the condensation it may be noted that such growth also occurs, although it is significantly weaker. Taking this into account, for example, when cooling with a water spray at a temperature of 21°C the heat transfer coefficients reach the values of the order of $10^5$ kW/(m²·K). The value $\sim 10^4$ W/m²·K cannot be regarded as highly exaggerated estimate for the heat transfer from the steam to the interface. For our problem, especially for low subcooling of liquid, steam generation at the interface can be substantial, and in this case, small variations in the pressure of steam will determine its motion. Since the project estimates do not look uniquely compelling, let us turn to some works, where the film boiling of subcooled water was examined. Much work on the analysis of film boiling on the surface of a sphere was performed by Liu et al. (1996). The authors discuss both the results of theoretical studies and a large quantity of experimental data obtained by them. An important limitation of the theoretical approach is the neglect of the vapor density compared to the density of water. This assumption is obviously unfair at high pressures. It also leads to the disregard of the dynamic effects at the phase boundary. The use of theoretical research results (Liu et al., 1996) does not promote the process description at high pressures ignoring convective heat transfer in the vapor phase and adding $0.5c_p\Delta T_{SUP}$ to the heat of the phase transition for taking superheat into account. This additive at high superheat and high pressure can be comparable with the heat of phase transition. Nevertheless, we will consider the results of Liu et al. (1996) which have a quite interesting character. Figure 4 illustrates the qualitative features of the process at low pressures. The dimensionless values used here are presented as follows:
FIG. 4: Experimental and calculated results of Liu et al. (1992) in the coordinates proposed by the authors for two subcoolings: 20°C and 30°C.

\[ C_{\text{sub},F} = \frac{\text{Nu}}{\text{Re}_l^{1/2}} \frac{\mu_l S_p'}{\mu_l \text{Sc}'} \left( \frac{1}{\text{Pr}_l^{1/2}} \right) \; \text{Sc}' = \left( \frac{c_{pv}\Delta T_{\text{sub}}}{r \text{ Pr}_v} \right) ; \]

\[ S_p' = \left( \frac{c_{pl}\Delta T_{\text{sub}}}{r \text{ Pr}_l} \right) ; \quad \text{Fr} = U_l^2 / (gd) , \]

where \( c_{pl} \) and \( c_{pv} \) are the heat capacities of liquid and vapor and \( r \) is the latent heat.
We shall also make some comment that will be useful later. It is easy to show that
\[
\frac{\mu v S_p'}{\mu_1 S_c'} = \frac{\lambda_f \Delta T_{sub}}{\lambda_v \Delta T_{sup}}.
\]  
Taking into account that the value $Pr^{0.5}$ for water at considered temperatures is close to 1 (1.32 at 100°C and decreases with temperature growth) in the expression for $C_{sub,F}$ the only remained quantities are the Nu and Re numbers. As can be seen from Fig. 4, heat transfer behaves as follows. At low Froude numbers, i.e., when the gravitational forces dominate, the heat transfer level is weakly dependent on the speed. For large Froude numbers its significant growth occurs. Let us try to use formally the results of Liu et al. (1996) for our conditions. If the jet diameter is about 1 cm and the speed is about 5 m/s, then the Fr number is greater than 10, the $Re_f$ number for our conditions is of order $4 \times 10^5$.

If we substitute the values of Fig. 4 in dimensionless coordinates, corresponding to the value of superheat ($\sim 160^\circ$C), for the heat transfer coefficient we obtain the value of order 50 kW/(m²·K). Most likely, this value is excessively high, but it still characterizes the growth trend of $\alpha$ on change of the regime parameters. Similar results were obtained by Lexin et al. (2009). Liu et al. (1996) together with the results of experimental studies also proposed a theoretical description of the process and theoretical expression for heat transfer coefficient. They are used to some degree in the approach that has been mentioned above with reference to Lexin et al. (2009), that is, the use of a joint description of boiling and condensation. In this part of the paper (Lexin et al., 2009), the mode of microbubble emission boiling was mentioned, which was investigated by several authors in connection with the analysis of nucleate boiling at high velocities and subcoolings.

There is a clear logic in the results of Liu et al. (1996) in reference to atmospheric pressure. However, attempts to use a high pressure do not allow one to obtain reliable results.

Taking into account that we need to develop a method of assessing the considered process for very high pressures, the use of the results cited above, unfortunately, cannot provide the required reliability. Attempts to find the literature data for experiments at a high pressure simulating the process of quenching hardly make sense. Now we consider some of the results of Liu et al. (1996) that can be used in the construction of the calculation method suitable for high pressures. In this work, the method of additive accounting for various factors that affect the intensity of heat transfer is used for creation of computing formulas. The constructed correlations for the Nusselt number include $Nu_p$, $Nu_v$, and $Nu_f$, i.e., three of the Nusselt numbers that correspond to pool boiling, flow boiling of saturated liquid and subcooled boiling. Since the whole theory is built for the atmospheric pressure, the pressure dependence in these numbers is, of course, absent. The lack of experimental data, which was mentioned above, does not allow one to use this method directly. But there is the following possibility. Data for a
saturated liquid can be obtained from experiments on post-CHF heat transfer and film boiling for a saturated liquid. Then we replace them with the help of $\text{Nu}_p$ and $\text{Nu}_s$ and thereafter determine corrections due to subcooling and use them in new dependences.

3. TEST DATA AND CORRELATIONS FOR POST-CHF HEAT TRANSFER

Before considering data for post-CHF heat transfer at high pressures, we will consider the results, which were obtained in the experiments of Chen et al. (1989) and are shown in Fig. 5.

In this work, devoted, in particular, to the study of heat transfer in inverted annular flow, experiments were conducted under laboratory conditions, but at pressures of up to 5.8 MPa. This differs from most of the papers mentioned above.

Interesting results were obtained concerning the effect of pressure on heat transfer. Unfortunately, the quality of this figure leaves much to be desired. Nevertheless, we decided to still insert it, given the curious character of the data that illustrate the possibility of reducing the heat transfer intensity on increase in pressure. As the author points out, the effect of reducing the heat transfer is shown in high subcooling. For this illustration, the author used data for a small mass velocity (by the way, other velocities are absent in this work) and for pressures less than 0.55 MPa. But when we analyzed experimental data for really high pressures, we could not discover such an effect.

FIG. 5: Experimental data adapted from Chen et al. (1989) demonstrates heat transfer decrease with pressure growth
It is worth noting that there is some problem in the use of data for post-CHF heat transfer. The fact is that the transition to practical studies of post-CHF regime in steam generating channels at actual mass velocities occurs at positive steam qualities. The reason is that in this case the values of the CHF are extremely high for subcooled water. But for positive void fraction, the liquid phase is at a temperature close to the saturation temperature, i.e., a high subcooling cannot take place. Taking into account the purpose of our research, it is desirable to use the data with minimum steam quality. Careful study of correlations, verified by the experimental data, showed that the minimum value of $X$ in the experiments was about 0.07. Thus for a pressure of 15 MPa the void fraction is of the order of 0.3 or slightly less. Therefore, the flow of liquid interacting with the steam can be regarded as a single phase flow. In this area, there is enough data for high pressures. Experimental formulas that were derived in the 60s–70s of the last century are as follows (Groeneveld, 1973):

$$\text{Nu}_v = a[\text{Re}_v \{X + \frac{\rho_l}{\rho_v} (1 - X)\}]^b \text{Pr}_w^c Y^d q^e \left(\frac{\rho_l}{\rho_f}\right)^f$$

or

$$\text{Nu}_l = a[\text{Re}_l \{X + \frac{\rho_l}{\rho_v} (1 - X)\}]^b \text{Pr}_w^c Y^d q^e \left(\frac{\rho_l}{\rho_f}\right)^f.$$  (3)

They, in principle, may contain additional factors. Their difference is that in the former case Nu and Re numbers are defined with the use of the physical properties of vapor, while in the second case — with the use of the physical properties of the liquid phase.

We have already mentioned the works with a minimum of void fraction of 0.07 (Bishop et al., 1964, 1965; Bennett et al., 1967; Groeneveld, 1972). In these studies the correlations were obtained in the ranges of pressure 4.08–21.9 MPa and mass velocities 700–3140 kg/m²·s that more or less correspond to our conditions. In the same range of pressures mass flow experiments were performed in the work of Miropol’sky (1963). However, the interval of steam quality was not specified here.

Nevertheless, both Miropol’sky’s formula and Bishop et al.’s formulas were used for evaluating heat transfer at a pressure of 15 MPa (Table 1). Estimations based on expressions (3) for a pressure of 15 MPa gave the following approximate results:

$$h \sim 20 \text{ kW/(m}^2\text{-K)} \ (\text{Bishop et al., 1964, 1965}) ;$$

$$h \sim 12 \text{ kW/(m}^2\text{-K)} \ (\text{Miropol’sky, 1963}).$$

Let us explain the meaning of the term "approximate" in this case. Of course, both formulas are not accurate enough, but that is not the point. As can be seen in the two used
formulas, the value of the surface overheating does not appear explicitly. In this case, it is replaced by the steam superheat $\Delta T_{\text{sup}}$. In the traditional method, this value was included in the expression for the heat transfer coefficient $h$. Hence, this effect should be kept in the change of the physical properties of steam with temperature growth.

We draw attention to the fact that the results obtained in Liu et al. (1996) and in a number of other works show that the strongest influence on heat transfer is exerted by the liquid phase subcooling. For an approximate estimate a multiplicative correction was proposed in the form of $1 + 0.2 \Delta T_{\text{sub}}$. If such an estimate is used for the heat transfer coefficient in comparison with the case of saturated liquid, then for $\Delta T_{\text{sub}} = 10^\circ\text{C}$ it is equal to three, for $\Delta T_{\text{sub}} = 20^\circ\text{C}$ to five, and so on.

This suggests that namely this temperature difference should be considered as a base for evaluating the heat transfer rate at high subcooling. This approach is considered in the next section, where a linear dependence is included into the formula used.

Given the relatively low accuracy of calculations mentioned above, let us present two examples showing both the possible errors and the possibility of adapting the results of calculations to real data by narrowing the consideration range. Figure 6 shows a comparison of the curve calculated from the data listed in the tables, for example, in Kirillov (2001) and the experimental data given in the same reference. As one can see, there is not only a quantitative but also a qualitative disagreement between the calculation and experiment. On the other hand, Fig. 7 presents the results in the field of the greatest interest to us in connection with this problem. However, here we can give another example that shows the possibility of a successful analysis of the inverted annular regime.

In the work of Agafonova and Paramonova (2013), devoted to the improvement of the Russian code KORSAR, a somewhat modernized variant of the formula, proposed in Hammouda et al. (1997) was used.

As a result, they successfully managed to describe the process of heat exchange in the area of small true steam qualities, which meets our conditions.

4. SIMPLEST THEORETICAL EVALUATION OF HEAT TRANSFER THROUGH THE WATER JET–STEAM INTERFACE

Before going to specific evaluation of heat transfer through the phase boundary, we note the following. In the foregoing the similarity of the processes occurring during

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<th>$a$</th>
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<td>0.8</td>
<td>0.8</td>
<td>1</td>
<td>0</td>
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<td>Miropol'sky, 1963</td>
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**FIG. 6:** Comparison of calculations by using the data listed in the tables with experimental data for a round tube: $p = 16$ MPa, $G = 1000$ kg/m$^2$·s, $q = 0.6$ MW/m$^2$. Lines, calculations; points, experimental data.

**FIG. 7:** The dependences of the heat transfer coefficient on the relative enthalpy at $p = 10$ MPa, $G = 4406$ kg/(m$^2$·s) experimental data adapted from Lappperiere and Groeneveld (1984), and $q_w = 496$ kW/m$^2$: 1) experimental data, 2) calculated results adapted from Agafonova and Paramonova (2013), and 3) calculated results of original model data adapted from Hammouda et al. (1997)
cooling after quenching and in the inverted annular mode was indicated. However, there is an important fact regarding the similarity. The point is that in these processes, superheated steam is generated by evaporation of a film when it is heated by steam conduction and convection and by emission from the heated solid surface. Therefore, there is no problem of the balance between the heat sink into the interior volume of the subcooled liquid from its surface in contact with the steam and the amount of steam cooled to the saturation temperature. The balance is achieved by itself due to the interaction of the heating layer surface and interface with steam. In the considered case, the superheated steam is fed into the space surrounding the jet, and its outside temperature and the flow rate are determined by external factors. When we considered this issue above, it has been shown that the process of superheat removal reaching the boundary is not actually determined in our case.

Therefore, in developing the model of the process both of these factors should be taken into account.

We now turn to the construction of the process models. Taking into account the estimated scale waves at the interface, we used the model to analyze the capillary waves. Let us now try to analyze the effect of these capillary waves on heat transfer to liquid and vapor phase boundary. Estimates made in Gotovsky et al. (2012) with the use of dimensional analysis related to heat transfer from the liquid to the phase boundary. Since saturated condensation was considered, the question about heat transfer to the steam in the process of condensation was not raised at all. As is well known (Levich, 1962), the capillary wavelength spectrum is occupied by \( l \), defined by the inequality

\[
l << 2\pi \sqrt{\frac{\sigma}{g \rho}} = n2\pi L_\sigma,
\]

where \( n \) is much smaller than unity.

The question is how to choose the interval for the wavelength. To eliminate the arbitrariness in this issue to some degree, the actual characteristics of the capillary waves were used, measured in Sadovsky et al. (2004) for optical studies of capillary waves. It should be noted that variations of this size have a little effect on the final result.

We will take into account the magnitude of the steam gap obtained above for \( p = 0.1 \) MPa and the previously selected \( n = 0.1 \), which is equal to 0.3 mm (3·10^{-4} m). Then \( l = 0.0016 \) m, the wave vector \( k = 2\pi/l = 4000 \) 1/m, and

\[
\omega = \sqrt{\frac{\sigma}{\rho'}} = 2000 \text{ 1/s}.
\]

We assume that the amplitude \( A \) is of the order of magnitude smaller than the wavelength. Then, for example, for atmospheric pressure we obtain that the full oscillation period is \( \tau = 2\pi/\omega = 3.14\cdot10^{-3} \) s and

\[
\text{International Journal of Energy for a Clean Environment}
\]
the average transverse velocity is \( U = 2A/\tau \approx 0.3 \text{ m/s} \).

Taking into account the sufficient roughness present, the model used can be called heuristic. As with any heuristic model, its success is mainly determined by its proximity to the results of estimating experimental data in this range of parameters.

Further the model of the process within the chosen method of description looks something like this. There are two recurrent phases:

1. The crest of the wave, containing superheated steam falls into the volume of subcooled water, wherein a layer of cooled steam formed close to the crest of the wave begins to condense on the surface of the water, creating a layer of water, whose temperature varies according to the thickness of the temperature to the saturation temperature of subcooled water. This process occurs within the half-period of oscillation.
2. At the same time, after the first half-period, the wave crest containing subcooled water enters the volume of superheated steam and the opposite process takes place. That is, a certain amount of water can evaporate.

It should be noted that the phase of the previous period of the process prepares the next phase. After the first phase, some amount of unstirred water reduces the heat transfer from the boundary, which promotes phase evaporation. After the first phase, the yet unstirred amount of water reduces the heat of the boundary, which promotes the evaporation phase. After the evaporation phase, an excess of warmed water is removed and the phase condensation proceeds more actively. Of course, the intensity of condensation substantially exceeds the intensity of evaporation, particularly at lower pressures.

It must be emphasized that each phase takes place within short periods of time for which the order of the numerical values is given above. Therefore, it is determined by the intensity of the transient heat conduction media. A detailed examination of this scheme is very complex, and it is most important that the greatest difficulty is posed by the very formulation of the problem. Therefore, we neglect the contribution of evaporation to the heat balance. Indeed, in contrast to the film boiling on a solid surface, in this case the superheated steam is fed into the system from an external source. It is obvious that the approach used has meaning only for sufficiently high liquid subcooling. Let us now try to estimate the quantitative values of heat transfer coefficients.

We assume that the heat transfer is limited only by the nonstationary heat conduction in the outer layer of the liquid phase. Taking into account that for the period of oscillation the heated layer thickness is small as compared to the amplitude of the wave, the formula for the time-averaged heat flux density on the surface of a flat layer at a constant temperature can be used.

Then after some transformations we obtain

\[
\frac{q}{\Delta T_{\text{sub}}} = A \lambda_l g^{1/4} L_{\sigma}^{-1/4} a^{-1/2} = \alpha_C, \quad (5)
\]
where \( q \) is the heat flux density which can absorb liquid during the contact period, as shown above, and over the oscillation period averaged and divided by the total wavelength. The coefficient \( A \) is approximately equal to 1.2.

The quantity on the right-hand side could be called the heat transfer coefficient, but usually the heat transfer coefficient in film boiling is attributed to the difference \( T_w - T_s \).

In our case, superheat \( \Delta T_{\text{sup}} \) will be used instead of this difference. Then we get

\[
\frac{q}{\Delta T_{\text{sup}}} = h = A\lambda/\sigma g^{1/4} L_{\sigma}^{-1/4} a^{-1/2} \frac{\Delta T_{\text{sub}}}{\Delta T_{\text{sup}}}. \tag{6}
\]

This shows the linearity of the \( h(\Delta T_{\text{sub}}) \) relationship. Here are the estimates obtained for the heat transfer coefficients. The main feature of the resulting formula is the practical absence of the dependence on pressure. Note that in the above formula there is no dependence on the mass velocity. The adopted approach uses a model of transient heat conduction.

But the model for integration of the data for the stream interacting with saturated steam comprises essentially the same approach which is based on the model of Levich (1962) Therefore, the possibility of taking into account the effect of velocity in the development of the model proposed here is real (Fig. 8).

The physical basis for this account is as follows. Since the proposed method corresponds to a sufficiently high velocity jet, it is necessary to quantitatively formulate this condition. A comparison with experimental data of Liu et al. (1996), where the

![FIG. 8: Scheme of heat transfer estimation for condensation of highly superheated steam on subcooled water jet](image)

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experimental results are presented in detail for a real jet diameter, this condition is satisfied when the water velocity (velocity of the jet) is 1–1.5 m/s, which corresponds to a positive slope of the curves in Fig. 4.

Since the velocity of the jet is usually much higher, the values given by the formula derived for heat transfer can be considered as minimal. The real values of the velocity can be corrected in the same manner as it was done in Liu et al. (1996).

Here we derived an equation in dimensionless form. To do this, we multiplied both sides by the diameter and divided by the thermal conductivity of steam. Then we obtained

\[
\frac{q}{\Delta T_{\text{sup}}} = \alpha = A\lambda_I g^{1/4} L_{\sigma}^{-1/4} a^{-1/2} \frac{\Delta T_{\text{sub}}}{\Delta T_{\text{sup}}^n}.
\]

(7)

This shows the linear character of the relationship.

Let us show the obtained estimates for the heat transfer coefficients \( h_C \).

The derived equation can be put in dimensionless form. To do this, we multiplied both sides by the diameter and divided by the thermal conductivity of steam. Then we obtained

\[
\frac{\nu}{\lambda_I g^{1/4} L_{\sigma}^{-1/4} \sigma} = \frac{\lambda_I \Delta T_{\text{sub}}}{\lambda_I \Delta T_{\text{sup}}}.
\]

(8)

In this formula \( d \) is the characteristic size, which in this case is the initial diameter of a jet, \( L_{\sigma} = \sqrt{\frac{a^2}{g}} \) can be called the thermal size, and \( L_{\nu} = \sqrt{\frac{v^2}{g}} \) is similar to the characteristic size which is used in processing the experimental data on condensation.

The last term on the right-hand side is similar to that introduced in Liu et al. (1996) and, as shown above, it replaces the expression for \( S_{\text{sub},F} \) [see Eq. (2)]. If we assume that at extremely high velocities \( \nu_c \) should tend to \( \nu_{\text{lim}} \) for the condensation of vapor on the jet, then the result can be expressed in the form

\[
\nu = \left[ \nu_c^n + (f \nu_{\text{lim}})^n \right]^{1/n}.
\]

(9)

The use of this formula is only possible at velocities providing a higher heat transfer coefficient than that considered here. The estimates show that the atmospheric pressure is the border for taking into account the impact of the transition rate corresponding to a Re number of \( 10^5 \).

It is also interesting to note that the value of the dynamic speed \( v^* \) for these conditions is in good agreement with the value of the transverse velocity fluctuations of the surface of 0.3 m/s mentioned above.

We introduce the notation \( y = \nu_{\text{lim}}/\nu_c \). According to the accepted condition, \( y \geq 1 \). The function \( f(y) = 1 - \exp\left(1 - y^2\right) \) provides a smooth transition between the regimes considered here and the limiting heat transfer to saturated steam. The quantity \( n \) can take a value of five.
Naturally it is surprising that such a simple approach is quite effective. There are significantly more complex models that, however, do not give sufficiently accurate results, despite the awkwardness of dependences. This usually happens when dependences include variables that do not affect the behavior of the system, but are significantly dependent on global variables (temperature, pressure, boundary conditions, etc.).

Now, given the nature of the problem, we also consider the situation with the external side of the steam. The simplest possibility is to use, for the first assessment, the approach similar to that for the liquid. We do this for the case of a high pressure. In connection with the interest associated with a new project of NPP with a BREST-300 reactor, we take the pressure to be equal to 16 MPa and the superheat to $\Delta T_{SUP} = 153^\circ C$. It is curious enough that under these conditions the heat of phase transition and enthalpy overheating are comparable with each other — 931 kJ/kg and 713 kJ/kg. Their ratio is 1.305. The estimated value of the heat transfer coefficient from the steam side in this case is 2.9 kW/(m$^2$·K).

The full enthalpy difference perceived by the jet is 1644 kJ/kg, and that supplied with the steam is 713 kJ/kg. The ratio of the specific streams supplying steam and perceived by the film $N$, provides thermal balance under the initial conditions of condensation:

$$N = 713/1644 = 0.434 .$$

On the other hand, this ratio for the superheat of 153$^\circ$C ($T_m = 500^\circ$C) is defined as

$$N_1 = (\alpha_v \Delta T_{sup})/(\alpha_C \Delta T_{sub}) = 2.9 \ 153/(18 \Delta T_{sub}) ,$$

where $\alpha_C = 18$ kW/m$^2$·K is obtained by extrapolation of the results listed in Table 2.

For example, $N_1 \approx 0.82 \ (\Delta T_{sub} = 30^\circ$C) and $N_1 \approx 0.62 \ (\Delta T_{sub} = 40^\circ$C).

Thus, at the chosen values of subcooling and superheat the calculations show an excessive heat supply to the steam side. Thus, there is a partial vaporization of the jet surface. That is, when the superheated steam should be cooled. But under other conditions the balance can be disrupted in the opposite direction. In that case, the balance must be established by changing the temperature field in the liquid phase.

In conclusion, there are several remarks on the topic discussed in the article. The topic is quite new, because direct interaction between a highly superheated steam and fed water practically has not been considered. There were doubts about the possibility of creating an efficient mixing heat exchanger with superheated steam. The development of a mixing high-pressure heater for NPPs with a BREST-OD-300 reactor of new generation required consideration in this problem. The All-Russian Institute of

<table>
<thead>
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<th>$P$, MPa</th>
<th>0.1</th>
<th>1</th>
<th>10</th>
</tr>
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<tbody>
<tr>
<td>$h_C$, kW/(m$^2$·K)</td>
<td>15.8</td>
<td>16.4</td>
<td>17.4</td>
</tr>
</tbody>
</table>
Thermal Engineering started these works in around 2005. Unfortunately, in the VTI model studied at the first stages of the operation, the process was divided into two stages: cooling of superheated steam with industrial water in surface heat exchangers and heating of the fed water with saturated wet steam (Somova et al., 2009a,b). Only at the end of 2016 the results of the study of a heat exchanger with superheated steam were published (Somova et al., 2015). True, the superheated steam temperature was reduced to 400°C. Tests of the model have shown that this idea can be fully realized.

A more detailed analysis of the problem will require the development of a more complicated model of the oscillating layer, as well as carrying out experimental verification.

5. CONCLUSIONS

The considered approximate scheme of the process of heat transfer through the phase boundary allowed us to understand the mechanism of substantial intensification of heat transfer in film boiling of a strongly subcooled liquid. It was possible, using a simple calculation method based on the formulas of nonstationary heat conduction, to get close enough to the experimental heat transfer coefficients.

Also important is the conclusion of no significant increase in heat transfer coefficient with increasing pressure.

Assessment of the feasibility of removing the superheating near to the phase boundary shows that a high-pressure re heater using superheated steam is quite real.

One can count on the fact that further development of the method of describing the processes of heat transfer at the interface with regard to its nonstationarity will significantly increase the level of quantitative description of heat transfer for consideration of specific cases in the given paper and for other cases of interaction between a subcooled liquid and superheated steam.

REFERENCES


