# Improving of thermohydraulic method for calculation of steam contact heat and mass exchange equipment

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## Abstract

**Introduction.** The complexity of determining the contact surface of phases and the boundaries of the continuous structure of liquid jets complicates the calculation of phase-contact heat and mass exchange mixing apparatuses.

**Materials and methods.** The condensation process of water steam from a steam-gas mixture on a cylindrical free-falling liquid jet is considered under counter-current movement of the steam phase within a range of flow parameters characteristic of the food industry.

**Results and discussion.** An empirical dependence for determining the onset of jet structure destruction adequately describes the process of its structural changes and is characterized by the fact that an increase in the Reynolds number leads to an increase in the critical height of jet destruction: the jet becomes more resistant to the action of the steam flow. With an increase in the dynamic pressure of the steam flow, corresponding to an increase in the Weber number, which characterizes a sharp decrease in the dimensionless height of dispersion, the jet intensively disintegrated, was destroyed at the outflow point, and was carried away by the steam flow.

An original system of dimensionless similarity numbers, based on the results of jet hydrodynamics analysis, allows for determining the temperature change along the jet's length, taking into account the geometric characteristics of mixing heat exchanger distribution devices, flow geometric dimensions, steam-liquid flow parameters, and the thermophysical properties of the media.

Empirical dependencies of the heat exchange process, considering the critical height of the existence of the continuous jet structure, are recommended for use in the thermohydraulic calculations of direct phase-contact heat exchange equipment.

**Conclusion.** The novelty of the results lies in the introduction of empirical dependencies for calculating heat exchange, considering the critical height of the existence of the continuous structure of the liquid jet.

## Introduction

Analyzing the most common calculation methods for steam-contact heat and mass transfer mixing equipment in the food industry (Bondar et al., 2015; Forsberg et al., 2020; Lienhard et al, 2024; Sokolenko et al., 2019), where the steam condenses on liquid jets, it was concluded that their use has certain limitations.

This is because any structural calculation of heat exchangers starts with a thermohydraulic calculation. The feature of these calculations is that when determining the heat exchange surface area, the total surface of the continuous jets is considered. It is also assumed that the liquid jet moving in the steam space has a continuous structure (does not break) throughout the heat exchange process and has a cylindrical shape. However, the continuity of the jet structure moving in the steam space depends on the parameters of the liquid and steam flows (Forsberg et al., 2020). Moreover, the liquid jet oscillates, and its outer surface shape differs from the ideal and does not match the shape of the orifice from which the outflow occurs (Bondar V., 2015). Consequently, it is quite difficult, and sometimes impossible, to calculate the cross-sectional area and, accordingly, the contact surface area between the liquid and the steam. Therefore, for these tasks, using classical equations, specifically the Newton-Richman equation (Kim et al, 2021), is not possible. The above implies that it is incorrect to use the traditional concept of the heat transfer coefficient, and thus it is proposed to use a dimensionless complex that describes the degree of temperature change of the liquid along the jet length under certain geometric conditions. Hydrodynamic regimes also determine the conditions for heat and mass energy transfer (Sokolenko et al., 2019).

Thus, using the currently existing classical methods for calculating mixing heat exchangers without considering the above is incorrect.

Research aim – to determine the limits of using the most common thermohydraulic calculation methods for heat and mass transfer equipment with direct phase contact used in the food industry.

## Materials and methods

Two-phase steam-liquid flows are studied. The process of water steam condensation from a steam-gas mixture on a cylindrical free-falling liquid jet with countercurrent steam flow.

Theoretical and experimental thermohydraulic study of the process of water steam condensation from a steam-gas mixture on a cylindrical free-falling liquid jet with countercurrent movement of the steam phase within the range of flow parameters typical for the food industry (Bondar et al., 2018).

In the first stage it was analyzed the main methods for calculating heat exchange during steam condensation on the surface of liquid jets. In the second stage it was analyzed the conditions under which the existing methods can be used. The experimental results were processed using statistical methods and mathematical modeling.

### Experimental installation

The experimental installation is designed to study the heat exchange process and determine the hydrodynamic characteristics during the condensation of steam on a cylindrical free-falling liquid jet over a wide range of flow rates and operating parameters of two-phase flows (Figure 1).

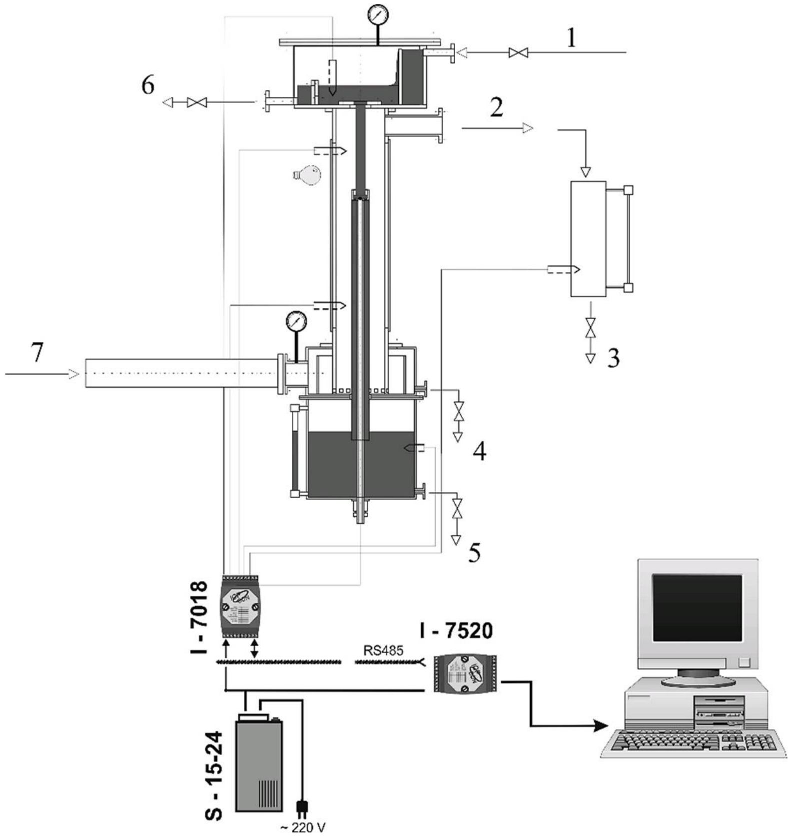


**Figure 1. Experimental setup diagram with directions of steam-liquid flows:**
1 – water supply from the network to the pressure tank;
2 – steam movement into the condenser;
3 – condensate drainage;
4 – drainage of parasitic condensate;
5, 6 – liquid drainage;
7 – steam supply to the experimental setup.

In the experiments, the initial parameters of the liquid and steam flows varied within the following ranges:

| Parameter | Denotation | Units | Limits of change |
|---|---|---|---|
| Heating steam pressure | $P_s$ | kPa | 101–114 |
| Steam speed | $v_s$ | m/s | 0–1.017 |
| Liquid speed | $v_0$ | m/s | 0.29–1.25 |
| Diameter of the orifice | $d_0$ | mm | 6, 8, 10 |
| Height of the liquid above the orifice | $h_0$ | mm | 60–155 |
| Flow rate of the liquid | $G_l$ | kg/s | 0.0015–0.0032 |
| Initial temperature of the liquid | $t_0$ | °C | 7–22 |

The water, on the surface of which the condensation process of water steam took place, enters the pressure tank from the network pipeline 1 through the inlet pipe. Then, the water stream flows out through a calibrated orifice under the action of gravity, enters the steam space filled with dry saturated water steam, and is heated by the heat of phase transition from the condensed water steam on its surface. The water steam enters the steam space above through the steam pipeline 7 from the electric steam boilers installed in a separate room. The heated water stream with condensate enters the liquid collector. The part of the steam that did not condense on the liquid jet enters the condenser through the pipeline 2.

During the hydrodynamic study, measurements of the geometric characteristics of the jets, their trajectories, and the onset of jet dispersion were carried out depending on both the speed of the steam flow and the flow rate of the liquid. Using visual observation and photography, the geometric characteristics of the liquid flow were determined. In the case of counter-current flow around the liquid jet by water steam, the hydrodynamic regimes of the process were determined.

During the heat exchange studies, measurements of the average heat content temperatures along the height of the jet were carried out. For this purpose, at the beginning of the experiment, a washer with a calibrated hole of a certain diameter was installed, the level of liquid above the center of the hole, and the steam pressure were determined. Depending on the diameter of the hole and the level of the liquid above it, the flow rate of the liquid in the setup changed. The level in the pressure vessel was kept constant using an overflow line. The flow rate and, consequently, the steam velocity and its pressure in the apparatus were regulated using regulating valves at the inlet and outlet of the setup. The average steam velocity in the experimental section was taken as the determining velocity of the steam.

When the liquid temperature and steam temperature remained constant, the experiment began. The experiments lasted until the liquid collector was completely filled. Intermediate purging of the experimental setup and draining of the parasitic condensate into the drain were carried out between the experiments.

## Results and discussion

In heat and mass transfer equipment used in the food industry, the main hydrodynamic regime for a single liquid jet is its continuous structure along the entire length (Sokolenko et al., 2019). Therefore, when determining empirical dependencies for calculating heat transfer during steam condensation on the surface of a cylindrical liquid jet, heat transfer in jets of this type is present. That is, when calculating heat transfer, only the continuous section of the jet should be considered.

However, a free-falling cylindrical jet has a complex and variable configuration along its height (Bondar et al., 2018). Even if it does not break, it constantly oscillates, and the shape of its cross-section changes, differing from the shape of the distributor orifice from which the outflow occurs.

During heat transfer calculations, which depend on the jet surface area, the energy transfer mechanism within it, and the temperature difference, it is necessary to account for the complex nature of the heat transfer intensity variation along the jet length. This is also evidenced by the experimentally determined temperature changes along the height of the jets, as shown in Figure 1.

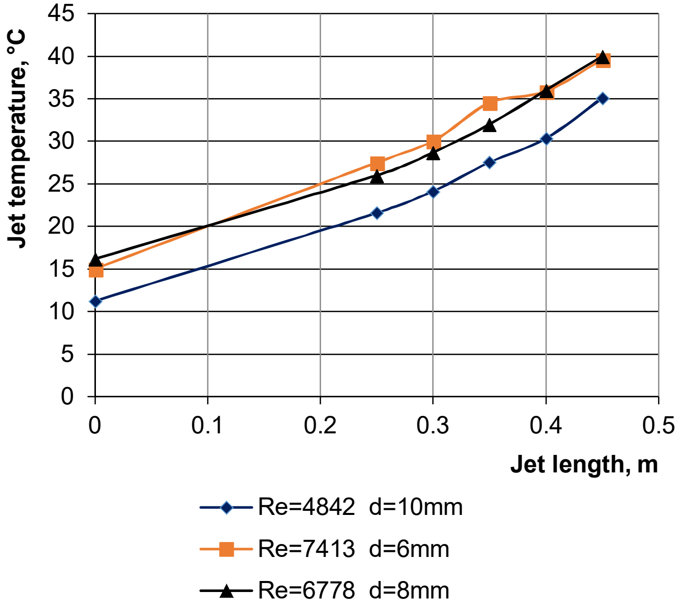


**Figure 1. Change in the average water temperature across the height of the jet**

According to the above, the use of existing methods for calculating heat transfer in cylindrical single liquid jets to analyze heat exchange in jets of this type is incorrect. This is evidenced by the results of comparing the data from this study with calculations based on existing correlations for both laminar and turbulent flow regimes (Bondar et al., 2018), which show qualitative and quantitative differences.

Under these conditions, it is impossible to construct an adequate mathematical model, and the experimental approach to obtaining an integral calculation method for heat exchange intensity becomes paramount.

When the liquid jet flows out, its shape does not correspond to the shape of the orifice, making it impossible to calculate the cross-sectional area and, consequently, the heat exchange surface area, rendering the use of the Newton-Richman equation is (Kim et al, 2021) impossible. This means that the traditional concept of the heat transfer coefficient cannot be used as a parameter, and instead, a dimensionless complex is proposed to describe the degree of temperature change of the liquid along the length of the jet under the corresponding geometric conditions:

$$4St = \frac{d_0}{y} \ln \frac{T_s - T_0}{T_s - T_p}, \tag{1}$$

where $T_s$ is the saturation temperature of the heating steam, K;

$T_p$ is the current temperature of the liquid, K;

$d_0$ is the orifice diameter, m.

In addition, considering previous studies on heat transfer during steam condensation on cylindrical jets and using the dimensional analysis method (Kovalenko et al., 2005), the following system of dimensionless parameters can be used in developing empirical correlations:

$$\mathrm{St}=f\left(\frac{1}{d_o};\ \mathrm{We};\ \mathrm{Re};\ \mathrm{K};\ \mathrm{Pr};\ \overline{\mu};\ \overline{\rho};\ \overline{\lambda};\ \overline{c_p}\right) \tag{2}$$

where:

$d_0$ – jet diameter, m;
1 – jet length, m;
St – Stanton number of the jet;
Re – Reynolds number of the jet;
Pr – Prandtl number of the liquid;
K – phase transformation criterion;
λ – thermal conductivity coefficient, W/(m·K);
μ – dynamic viscosity coefficient, Pa·s;
ρ – density, kg/m³;
ν – kinematic viscosity coefficient, m²/s;
v – jet velocity at the distributor cut, m/s;
$c_p$ – liquid's heat capacity, J/kg·K.

In processing experimental data, the last four parameters were not considered because the physical properties of the liquid and steam did not change significantly.

The phase transformation criterion (K) accounts for the release of heat from supercooled condensate and the hydrodynamic effects associated with the presence of transverse mass flow on the phase boundary surface.

The selection of determining parameters should consider the influence of characteristics of steam-jet flows, jet length, and the physical properties of the phases.

Analyzing the results of experimental studies within the dimensionless parameter systems (Bondar et al., 2019; Kim et al., 1989), it can be concluded that there is currently no universal system of parameters that can adequately describe the heat transfer process during steam condensation on a cylindrical free-falling liquid jet with sufficient accuracy. Therefore, an original system of dimensionless similarity numbers was developed based on the results of hydrodynamic jet analysis.

According to this system, for one-dimensional modeling, the experimental research results are approximated by the empirical dependency:

$$\frac{d_0}{y}\ln\frac{T_s - T_0}{T_s - T_p} = 0{,}000897\cdot\left(\frac{1}{d_0}\right)^{-0{,}73}\cdot \mathrm{Re}_0^{\;0{,}99}\cdot We_{\sigma 2}^{-0{,}53}\cdot K^{-1{,}66}\cdot \mathrm{Pr}^{-2{,}22} \tag{3}$$

where:

$d_0$ – jet diameter, m;
1 – jet length, m;
$T_p$ – liquid temperature, K;
$T_0$ – liquid temperature at the distributor cut, K;
$T_s$ – steam saturation temperature, K;
$We_{\sigma 2}$ – modified Weber number,
$Re_0$ – Reynolds number of the jet,
Pr – Prandtl number of the liquid;
K – phase transformation criterion;
λ – thermal conductivity coefficient, W/(m·K);
ν – kinematic viscosity coefficient, m²/s;
ρ – density, kg/m³;
σ – surface tension coefficient, N/m.

Furthermore, to describe the onset of dispersion (the transition boundary from wave to dispersed hydrodynamic regime (Bondar et al., 2015)), it is necessary to determine the dimensional quantities that define this transition. A preliminary analysis of the hydrodynamic study of a single jet movement in counter-current flow with water steam concluded that the transition between hydrodynamic regimes depends on the orifice diameter, liquid velocity, dynamic pressure of the steam flow, and the physical properties of the liquid and steam flows. Thus, the height at which jet dispersion begins depends on the following quantities: the liquid jet diameter, the kinetic energy of the steam flow, the liquid exit velocity from the orifice, viscosity, surface tension, and free-fall acceleration.

$$h_{cr} = f((\rho_p v^2{}_p);\ v_p;\ d_0;\ \nu_p;\ \sigma_p;\ \rho_p;\ g) \tag{4}$$

The onset of jet structure breakdown is proposed to be described by the following relationship of dimensionless parameters:

$$h^*_{cr} = f[\,We_{\sigma 1};\ Re], \tag{5}$$

where

$$h^*_{cr} = \frac{h_{cr}}{\sqrt{\sigma_p/(\rho_p \cdot g)}}$$

is the dimensionless vertical distance from the orifice center to the plane where jet dispersion begins;

$We_{\sigma 1} = \dfrac{\rho_p v_p^2}{\sqrt{\sigma_p \cdot \rho_p \cdot g}}$ is the Weber capillary number;

Re is the Reynolds number of the jet;

$$Re_0 = \frac{v_o \cdot d_o}{\nu_p}$$

where $v_o$ is the liquid velocity at the orifice exit, m/s;

$v_p$ is the steam velocity in the steam space, m/s;

$h_{cr}$ is the height at which dispersion begins, measured from the point of outflow, m;

$d_o$ is the orifice diameter, m.

It is also necessary to consider the change in the ical parameters of the liquid and steam flows by introducing a dimensionless complex, such as the Prandtl number (Pr). Flow parameters for constructing dimensionless complexes were selected based on practical considerations. In heat exchange equipment, it is possible to measure the orifice diameter rather than the jet diameter and calculate the liquid and steam flow velocities at the orifice exit.

The experimental data are satisfactorily approximated by the dependence (Bondar, 2015):

$$h^*_{cr} = 14{,}10840 \cdot Re_0^{\,0{,}180938} \cdot \exp\left(-55{,}54866 \cdot We_{\sigma 1}\right). \tag{6}$$

A comparison of experimental and calculated values using the empirical dependence (6) is shown in Figure 2.

The proposed dependence accurately adjusts the experimental data within a 10% error margin, with a determination coefficient of 98.11%.

To further verify the proposed empirical dependence, we will examine how the dimensionless height of jet dispersion onset changes with flow parameters. As previously discussed, flow parameters are accounted for by the dimensionless complexes $Re_0$ and $We_{\sigma 1}$. The comparison results are shown in Figure 3.

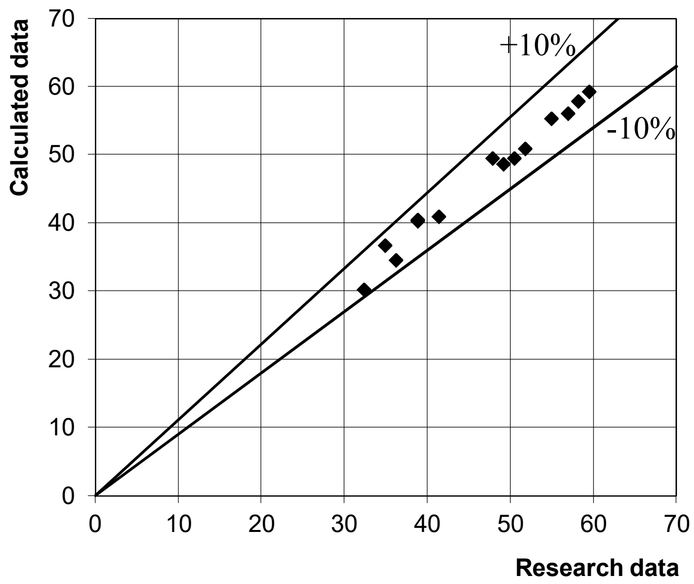


**Figure 2. Comparison of calculated and experimental values of $h^*_{cr}$**

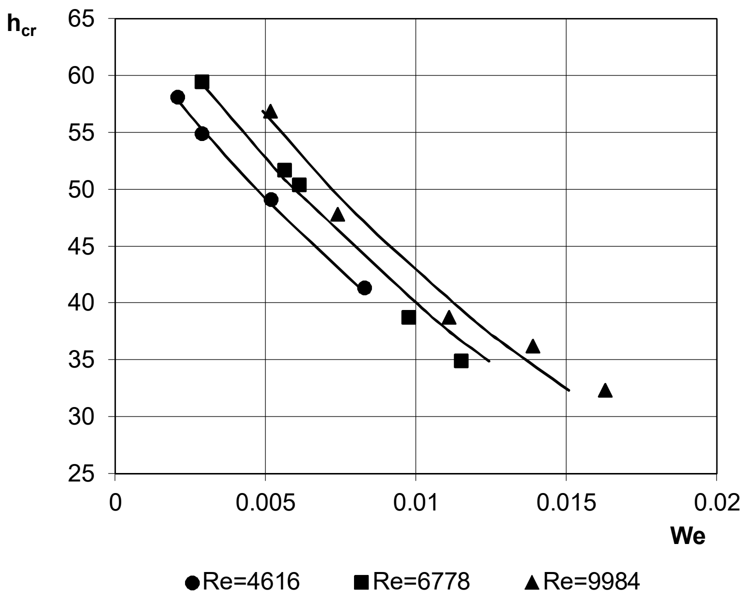


**Figure 3. Effect of the $We_{\sigma 1}$ criterion on the dimensionless height of dispersion onset.**
Calculated data by formula (6) are shown as lines, and experimental data are shown as points for corresponding Re values.

Analyzing the graphical dependence, we conclude that the experimental values systematically vary by Reynolds number. An increase in Re causes an increase in $h^*_{cr}$, meaning the jet becomes more stable against the action of the steam flow.

As the dynamic pressure of the steam flow increases and thus $We_{\sigma 1}$ rises, there is an initial sharp decrease in the dimensionless height of dispersion, followed by intensive jet fragmentation and destruction at the outflow point, leading to it being carried away by the passing steam.

We can conclude that the empirical dependence for determining the onset of jet structure breakdown adequately describes the process of its structural change.

Summarizing the above, we propose using the research results when designing new heat exchange equipment with direct phase contact.

Currently, there are many methods for calculating mixing heat exchangers. For example, the method of calculating a barometric mixing condenser is proposed to be improved, which consists of the following steps:

**1.** Determining the cooling water flow rate from the heat balance equation:

$$G_w = W\frac{\left(h_s - c_w \cdot t_{cond}\right)}{c_w \cdot \left(t_2 - t_1\right)}, \tag{7}$$

where W is the steam flow rate entering the condenser, kg/s;

$h_s$ – the specific enthalpy of the steam, kJ/kg;

$c_w$ – the specific heat capacity of water, kJ/kg·K;

$t_{cond} = t_2$ is the temperature of the condensate, °C;

$t_1$ and $t_2$ are the initial and final temperatures of the cooling water, °C.

**2.** Determining the diameter and height of the barometric condenser. The condenser diameter is determined from the flow rate equation:

$$D_{b.c.} = \sqrt{4W\upsilon / \left(\pi\omega_s\right)}, \tag{8}$$

where $\upsilon$ – the specific volume of the steam, m³/kg;

$\omega_s$ – the steam velocity in the condenser, m/s.

The obtained value $D_{b.c.}$. is rounded to the standard, based on which all dimensions of the condenser are selected.

A simplified calculation of the number of trays and the height of the condenser was conducted by assuming that the distance between trays is the same for all trays, and the amount of liquid dripping from each tray is equal $G_w$+W

The height of the condenser is determined by the formula:

$$H = \left(n+1\right)h, \tag{9}$$

where n is the number of trays,

h is the distance between trays, which is chosen within the range of 350–550 mm.

The number of trays is determined by the formula:

$$n = \ln\frac{T_s - T_1}{T_s - T_2} / \frac{T_s - T_0}{T_s - T_p}, \tag{10}$$

where $T_s$ – the saturation temperature of the steam, ˚C,

$T_1$, $T_2$ – the initial and final temperatures of the cooling liquid, ˚C.

Knowing the flow rate of the liquid, we assume that the fraction of liquid dripping through the cylindrical openings is equal to $0,3\left(G_w + W\right)$.

**3.** It is checked the condition of the jet continuity:

$$h_{cr}^{*} = 14,108 \cdot \mathrm{Re}_0^{0,181} \cdot \exp\left(-55,55 \cdot \mathrm{We}_{\sigma 1}\right)., \tag{11}$$

where $h_{cr}^{*} = \dfrac{h_{cr}}{\sqrt{\sigma_p / \left(\rho_p \cdot g\right)}}$,

$h_{cr}$ – the height at the start of dispersion;

$h_{cr} = h$, м;

If this condition is not met, the jet breaks down, and a new distance between trays is set.

**4.** The heat transfer on cylindrical liquid jets was calculated using the empirical dependence:

$$\frac{d_0}{y} \ln \frac{T_s - T_0}{T_s - T_p} = 0,000897 \cdot \left(\frac{1}{d_0}\right)^{-0,73} \cdot \mathrm{Re}_0^{0,99} \cdot We_{\sigma 2}^{-0,53} \cdot K^{-1,66} \cdot \mathrm{Pr}^{-2,22} \tag{12}$$

**5.** The heat transfer for the liquid flowing out through a slit distributor is calculated using the dependence (Vasylenko, 2003):

$$\frac{d_e}{y} \ln \frac{T_s - T_0}{T_s - T_p} = 8,59 \cdot 10^{-17} \cdot \left(y^{*}\right)^{-0,849} \cdot \mathrm{We}^{-0,023} \cdot \mathrm{Pr}_p^{-3,661} \cdot \mathrm{Re}_{de}^{4,108} \tag{13}$$

where $y^{*}$ – is the dimensionless distance from the water overflow edge,

$$y^{*} = y / \left[\sigma_p / \left(\rho_p\, g\right)\right]^{0.5}$$

$d_e$ is the equivalent diameter of a single jet, м;

$$d_e = 4.b.h/(2h + b);$$

b is the width of the slit distribution device, m;

$Re_{de}$ is the Reynolds number of the jet flowing from the slit,

$$Re_{de} = v_{0j}.d_e/\nu_p$$

We is the Weber number,

$$We = \rho_p \cdot v_p^2 \cdot d_e/\sigma;$$

b – width of the distribution device, m.

**6.** At the end of the calculation, a check is made to ensure that the liquid jet heats up to the set temperature, which flows through the slit and cylindrical distribution devices to the same temperature. If not, a new distance between trays is set.

Further calculations of the dimensions of the barometric pipe and the calculation of the vacuum pump are carried out using classical methods.

## Conclusions

1. When it is calculating the heat transfer in free cylindrical liquid jets during the condensation on their surface of stationary steam and steam from the countercurrent

flow, it is recommended to take into account the limits of the existence of a continuous structure of the jet.

2. It is recommended to calculate the heat exchange according to the dependencies given in the article, the adequacy of which is confirmed by comparative analysis.
3. A scientifically substantiated methodology for the thermohydrodynamic calculation of steam-contact heat and mass transfer devices has been developed. The feature of methodology the feature of methodology is that before calculating the heat transfer, the condition of the existence of a solid jet structure is checked, and the heat transfer is calculated according to the obtained empirical formula.

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