View metadata, citation and similar papers at core.ac.uk brought to you by CORE



UDC 536.248.2
DOI: 10.32626/2308-5916.2019-20.104-113

**I. E. Furtat,** PhD, Associate Professor

National Technical University of Ukraine
«Igor Sikorsky Kyiv Polytechnic Institute», Kyiv

# CONDENSATION HEAT AND MASS TRANSFER AT DIRECT CONTACT OF THE REACTING PHASES

Condensation upon direct contact of the phases can be divided into the following types: condensation of the steam stream in the volume of unheated liquid; condensation of vapor bubbles in liquid; condensation of steam by liquid droplets (dispersed liquid); vapor condensation on a jet of liquid.

In visual experiments, the study of the process of condensation of the jet of steam in space was noted by the presence of a white emulsion at the collision of steam with liquid, due to the crushing of a jet of steam into small bubbles. The high intensity of the heat transfer process was explained by the sharp increase in the contact surface. When considering the structure of the flow taking into account the two-phase region, it can be noted that there is a smooth conical surface of the separation between the phases and the formation of dispersed bubbles and droplets in the flow. This allows to determine the dependence of the geometry of the contact zone of the phases on the temperature head.

An increase in the surface area of the contact phase can be achieved by dispersing one of the contacting phases. Existing liquid spraying machines have significant energy costs as a result of doing some work to overcome the surface tension that causes the liquid to reduce the free surface.So the heat transfer between a liquid drop and a saturated vapor is determined by the heat distribution along the drop radius. The vapor condenses on the surface of the liquid droplet, and the released heat condensation must be discharged inside the droplet. According to the equation of thermal conductivity under the relevant conditions of the problem under consideration, the intensity of condensation is determined by the rate of heat runoff per drop. Studies of heat exchange on dispersed jets of liquid have proven high intensity of the process.

Condensation on a jet of liquid is used in many industrial devices (deaerators, condensers of mixing type, jet heaters).

Theoretical and experimental studies of this type of condensation are scarce. Studies of heat exchange during condensation of a dispersed steam stream on a swirling stream of water are absent at all.

The results of the experiments of heat exchange at the contact condensation of steam on jets of water, consisting of a continuous section and a section that falls into drops, are represented by the

 © I. E. Furtat, 2019

criterion equation. Recent studies are related to the development of a mathematical model for the calculation of jet condensation and analysis of past developments with its application.

**Key words:** *condensation, steam, liquid, jet, phase contact, heat exchange.*

**Introduction.** By analyzing the condensation in direct contact of the phases, it can be divided into the following types:

- condensation of steam stream in the volume of underheated liquid;
- condensation of vapor bubbles in the liquid;
- condensation of steam on liquid droplets (dispersed liquid);
- condensation of steam on a liquid jet.

Note that during the condensation the non-condensing gas contained in vapor leads to a decrease in the heat transfer coefficient due to the additional diffusional thermal resistance.

The first studies of the condensation process of a steam jet in a liquid filled space are based on visual experiments [1; 2]. The presence of a white emulsion was noted when the vapor collides with the liquid, due to the crushing of a steam jet into small bubbles. The high intensity of the heat transfer process was explained by the sharp increase in the contact surface.

**Earlier researches of the steam penetration into water.** As stated in [3], the penetration of steam into water due to the difference of partial pressures of steam in the vapor medium and in the liquid causes the leakage of steam at high speed, which explains the presence of the emulsion. Studies [4; 5] determined the theoretically obtained and experimentally confirmed position of the condensation surface depending on the parameters of the vapor and the liquid. The condensation process is said to end on the surface of the «condensation cone». This is due to the high intensity of turbulent mixing. On the basis of previous studies, [6] obtained a relation that determines the position of the condensation surface, and presents the dependence of the change in the full angle of the «condensation cone» $\beta_0$ depending on the temperature of the liquid $t_{liq}$.

Later publications [7–9] consider the structure of the flow taking into account the two-phase region. The presence of a smooth conical surface between the phases and the formation of dispersed bubbles and droplets in the flow are noted. The result of these recent studies [9] is to determine the dependence of the phase contact zone geometry on the temperature head $\Delta t$. The condensation process ends at the height of the contact zone, which is equal to the penetration depth $h$ of the steam stream. In addition, the condensation cone pulsations were noted [8; 9], as a consequence of the use of stabilizing screens in heaters of the mixing type. According to [10], the heat transfer equation has the form:

$$Nu = 6,5 \cdot \mathrm{Re}^{0,6} \cdot Pr'. \tag{1}$$

Describing the features of the mechanism of vapor bubbles condensation in liquid volume, one distinguishes the first stage of bubbles collapsing (large radius bubble), which is due to the intensity of heat exchange, and the second stage — (small radius) with the influence of inertial forces [11].

In publications [12] analytical and experimental material was obtained, characterizing the condensation of fixed steam bubbles. The intensification of the destruction of steam bubbles under the influence of their translational motion has been studied [13–16].

**Models of steam-water interaction with condensation.** Under conditions of mass bubbling [17], with the number $N_з$ of steam jets, the studied temperature fields along the length and height of the two-phase layer are exponential in nature. The formula of the coefficient of heat transfer from steam to liquid is obtained:

$$\alpha = 6,05 \cdot 10^3 \cdot C'_p \cdot \rho' \cdot \sqrt{\frac{a' \cdot U_0}{N_з \cdot L_\varphi}}, \tag{2}$$

where $C'_p$ is the specific heat of the liquid; $\rho'$ — liquid density; $a'$ — coefficient of thermal conductivity of the liquid; $L$ — length of free run of the bubble (jet); $\varphi$ is the steam content of the layer.

In [18; 19], a model of periodic renewal of turbulent surface moles is used to calculate the intensity of heat transfer through an interfacial surface, and a universal dependence is obtained, which determines the coefficient of interphase heat exchange in a turbulent flow:

$$Nu = 0,23 \cdot \mathrm{Re}^{0,7} \cdot \mathrm{Pr}^{0,5} \cdot A^{0,25}, \tag{3}$$

where $A$ is the correction that characterizes the steam content in the stream.

Formula (3) is applicable to sufficiently large bubbles in the region $\mathrm{Pr} \geq 0.3 \div 0.5$; $A \leq 0.04 \cdot (d_n/D) \cdot \mathrm{Re}^{0.7} \cdot \mathrm{Pr}^{0.5}$.

It is known that there are no experiments, calculations for the detection of theoretical laws of heat and mass transfer when mixing a dispersed jet of steam with a jet of liquid. This process is complex enough, so there is a need for experimental studies. An increase in the surface area of the phases contact can be achieved by dispersing one of the contacting phases. Existing liquid spraying machines have significant energy costs as a result of doing some work to overcome the surface tension that causes the liauid to reduce the free surface. It is known that for spraying liquid energy consumption of most existing dispersants is in the range of 3.5 ÷ 10 (kW·h)/t. Liquid dispersion also has the disadvantage of having an additional thermal drop resistance. So the heat transfer between the liquid droplet and the saturated vapor is determined by the heat distribution along the droplet radius. The vapor condenses on the surface of the liquid droplet, and the released heat of condensation must be discharged inside the droplet. According to the equation of thermal conductivity under the relevant conditions of the problem under consideration, the intensity of condensation is determined by the rate of heat runoff per drop.

The coefficient of heat transfer from steam to a drop of liquid [10] is determined by the formula:

$$Nu = 2 + 0,74 \cdot \mathrm{Re}^{0,7} \cdot \mathrm{Pr}^{0,33} . \qquad (4)$$

Studies [20] of heat transfer on dispersed jets of liquid have proven the high intensity of the process and have recommended the following formulas:

$$\frac{Q_v(x)}{r \cdot G} = 1,75 \cdot \mathrm{Re}_{03,n}^{0,33} \cdot X^{-0,27} \cdot K^{-1} \cdot \left[ 1 - \exp\left( -\frac{\pi^2}{Pe_{03,liq}} \right) \right], \qquad (5)$$

where $Q_v(x)$ is the local density of heat, *W*; *G* — mass flow rate of liquid, kg/s; $K$ = 7.62 ÷ 17.80; Re $_{03,\ n}$ = 2.0 ÷ 167.0; Pe $_{03,liq}$ = 2740 ÷ 15800; $X$ = 6 ÷ 412.

The experimental data is described by the equation:

$$\Theta = 1,25 \cdot 10^{-4} \cdot Lp^{0,62} \cdot K^{-0,45} \cdot A^{-0,59} \cdot \varepsilon^{-0,08} \cdot We^{0,02} \cdot Z^{0,03} \cdot X^{0,26}, \quad (6)$$

when changing values are within: $We$ = 6.6 ÷ 28.6; $Lp$ = (6.6 ÷ 29.3)·$10^5$ ; $K$ = 7.6 ÷ 18.7; $A$ = 1.13 ÷ 1.70; $\varepsilon$ = (0.1 ÷ 84.0)·$10^{-4}$ ; $Z$ = 0.42 ÷ 9.37; $X$ = 4.65 ÷ 47.40; while $\Theta$ = 0.20 ÷ 0.98.

Condensation on the jet of liquid is used in many industrial devices (deaerators, condensers of mixing type, jet heaters).

Theoretical and experimental studies of this type of condensation are scarce. The recommended calculation formulas for estimating the heat transfer during condensation of steam on a liquid jet differ by the methods of determining the basic parameters and components.

In experimental studies of I. V. Vasilyev in CCTU [21] a formula is obtained when water flows out of holes with diameter $d$ = 3 ÷ 7 mm ; jet height $H$ = 0.2 ÷ 0.55 m; initial velocity of water $w$ = 0.2÷1.4 m/s and temperature $t$ = 20 ÷ 90 °C:

$$\lg \frac{t_s - t_1}{t_2 - t_1} = 0,029 \cdot \left( \frac{g \cdot d_1}{w^2} \right)^{0,2} \cdot \left( \frac{H}{d} \right)^{0,7}, \qquad (7)$$

at Re > 1500; 1 < Fr = $\frac{g \cdot d_1}{w^2}$ < 40; 40 < $\frac{H}{d}$ < 100.

The formula theoretically obtained in [22] to determine the heat transfer coefficient has the form:

$$\alpha = 377 \cdot C_{p\,рід} \cdot \rho_{рід} \cdot w_0 \cdot \Phi(K), \qquad (8)$$

where $\Phi(K)$ is the function of the parameter $K = r / \left[ C_{p\,рід} \cdot (t_н - t_0) \right]$.

The theoretical formulas are proposed in [23; 24] according to the hypothesis that the coefficient of turbulence is proportional to the cross section and the absolute flow velocity:

$$\bar{\Theta} = \sum_{i=1}^{\infty} \frac{4}{\beta_i^2} \cdot \exp\left(-\beta_i^2 \cdot f(x)\right),$$
$$f(x) = \frac{4 \cdot a \cdot x}{w_0 \cdot d_0^2} + \frac{4 \cdot \varepsilon_* \cdot w_0^2}{5 \cdot \varphi^{2,5} \cdot g \cdot d_0} \cdot \left[\left(1 + \frac{2 \cdot \varphi^2 \cdot g \cdot x}{w_0^2}\right)^{1,25} - 1\right], \tag{9}$$

where $\varphi$ is the factor of the jet narrowing; $\varepsilon_*$ = 0.0005; $\bar{\Theta}$ — relative underheating of the jet.

**Studying of the jet heaters.** Valuable experimental studies of the external thermal and hydraulic characteristics of the jet heater were conducted in VTI [25; 26]. The effect on the heat exchanger design and mode factors was investigated. It is known that the shape of the nozzle does not affect the water heating and pressure value renewal $\Delta p_c$. The operation of the jet heater under the influence of mode factors was analyzed depending on the flow of working water, water temperature, injected steam pressure. In [28], the dependences of the underheating of water on the saturation temperature and the heating steam on the jet length are given. They prove that at high flow rates, the main heating occurs at a short initial jet section, at lower ones — over the entire jet length. The experiments were conducted using the formulas given above.

On the basis of theoretical developments [22], the process of condensation at the outflow of a cold jet of liquid with velocity $w_0$ from an aperture of radius $R$ into the vapor space was studied in [27] with the assumption that «the molecular heat flow is less intense than turbulent». The dependence of the dimensionless temperature Θ and the criterion $St$ on the geometric relation $\zeta = x / R$ and the dynamic parameter $\alpha_d$ is thus obtained:

$$St = f(\alpha_\partial) = f\left(\frac{2 \cdot g \cdot R}{w_0^2}\right), \tag{10}$$

where $St = \dfrac{\bar{\alpha}}{C_{p\,liq} \cdot \rho_{liq} \cdot w_0}$.

Experimental studies [28] for water jets in the transverse stream of steam in the region $p$ = 1÷100 kPa; $w_0$ = 0.8÷1.7 m/s; $\rho_n \cdot w_n^2$ = 4÷60 kPa; $d$ = 2÷15 mm ; $l$ = 0.2 ÷ 0.5 m are described by the criterion equation:

$$\lg \frac{t_н - t_0}{t_н - \bar{t}_1} = 0,085 \cdot \frac{l}{d} \cdot Lap^{0,33} \cdot K^{-0,13} \cdot \mathrm{Pr}^{-0,62} \cdot Fr^{-0,33} \cdot (1 - \pi)^2, \tag{11}$$

where $Lap = \dfrac{\rho \cdot w_n^2 \cdot d}{\sigma}$ is the Laplace criterion; $\pi = \dfrac{G_{air}}{G_{mix}}$ — the ratio of the mass of air to the total amount of mixture of air and steam.

Experimental studies [29] of the operation of the jet capacitor were performed depending on the design features of the apparatus and the operating conditions. Later, experimental data [30] for vertical jets in vacuum conditions are described by the dependence:

$$Nu = 0,02 \cdot m \cdot \mathrm{Re}^{1,2} \cdot \mathrm{Pr}^{0,43} \cdot K^{0,1} \cdot \left(\frac{l}{d_0}\right)^{-0,75}, \quad (12)$$

where $m = \frac{\Delta t_{avg}}{\Delta t_{\log}}$; under conditions $p$ = 15 ÷ 100 kPa; $w_0$ = 9 ÷ 26 m/s; $w_n$ = 10 ÷ 30 m/s; $d_0$ = 3 ÷ 20 mm; $l$ = 0.2 ÷ 1.2 m; number of jets 1 ÷ 46.

The results of the experiments of heat exchange at contact condensation of steam on jets of water, consisting of a solid section and a section that splits into droplets, [31] are represented by the criterion equation:

$$Nu = 2,7 \cdot \mathrm{Re}_{liq}^{0,6} \cdot \mathrm{Pr}_{liq}^{0,45} \cdot K^{0,11} \cdot We^{0,4} \cdot \left(\frac{l}{d}\right)^{-0,75}, \quad (13)$$

in the range $\mathrm{Re}_{liq}$ = (1,4 ÷ 9,0)·$10^5$ ; $\mathrm{Pr}_{liq}$ = 2.5÷4.0; $K$ = 9÷60; $We = 1.6 \cdot 10^2 \div 3.0 \cdot 10^3$ ; $l / d$ = 12 ÷ 60.

In the conditions of cross-motion of coolants for a large number of nozzles from which water flows, the formula is obtained:

$$\overline{Nu} = 1,57 \cdot Gz_{liq}^{0,5} \cdot K^{0,5} \cdot We_n^{0,28} \cdot \left(\frac{s_2}{d}\right)^{0,75}, \quad (14)$$

where $Gz = \frac{C \cdot z_{liq} \cdot \gamma_{liq} \cdot w_{liq} \cdot d^2}{\lambda_{liq} \cdot l}$ is the Gretz criterion ; $\frac{s_2}{d}$ — relative pitch of the jet.

The total surface of the condensation was equal to the surface of a single jet of liquid (lateral surface of a cylinder with a nozzle diameter $d_s$ and height equal to the length of the jet, l) multiplied by the total number of jets.

Relative heating Θ of a laminar jet in liquid [32] depends on the Fourier number $Fo$ and dimensionless radius $\varepsilon_i$:

$$\bar{\Theta} = \sum_{i=1}^{\infty} \frac{4}{\varepsilon_i^2} \cdot \exp\left(-\varepsilon_i^2 \cdot Fo\right), \quad (15)$$

where $Fo = \frac{0,5 \cdot \varepsilon \cdot l}{d}$.

From the thermal balance of the jet it was obtained that:

$$4 \cdot St = \frac{d_0}{l} \cdot \ln \frac{1}{\Theta}. \quad (16)$$

In the following experiments, the dependences of heat exchange for a continuous jet flowing from the nozzle from top to bottom into a large volume of steam are described [33]:

$$4 \cdot \overline{St} = 0{,}134 \cdot \left( \frac{l}{d_c} \right)^{-0{,}42} \cdot \mathrm{Re}^{-0{,}17} \cdot \mathrm{Pr}^{-0{,}09} \cdot K^{0{,}13} \cdot We^{0{,}35}, \qquad (17)$$

$$4 \cdot \overline{St} = 0{,}133 \cdot \left( \frac{l}{d_c} \right)^{-0{,}41} \cdot \mathrm{Re}^{-0{,}18} \cdot \mathrm{Pr}^{0{,}05} \cdot K^{0{,}11} \exp\left(0{,}16 \cdot We\right), \qquad (18)$$

where $\frac{l}{d_c}$ = 4÷180; Re = (1.5 ÷ 10.0)·$10^4$ ; Pr = 1.8 ÷ 6.4; $K$ = 6 ÷ 50; $We$ = 0.4 ÷ 5.5.

Equation (17) is recommended for $We \geq 2.7$ and equation (18) for $We \leq 2.7$.

The influence of the criterion $We$ on the heat exchange is revealed [34]:

$$\text{at } \frac{l}{d} \leq 95 \quad 4 \cdot St = 5{,}15 \cdot 10^{-2} \exp\left(0{,}135 \cdot We\right) \cdot \left( \frac{l}{d} \right)^{-0{,}54}, \qquad (19)$$

$$\text{at } \frac{l}{d} \geq 95 \quad 4 \cdot St = 1{,}5 \cdot 10^{-2} \exp\left(0{,}135 \cdot We\right) \cdot \left( \frac{l}{d} \right)^{-0{,}27} \qquad (20)$$

for Re = (1.5 ÷ 10.0)·$10^4$; $We$ = 1.1 ÷ 4.1; $k$ = (4.2 ÷ 17.4)·$10^{-2}$; $\frac{l}{d_c}$ = 12 ÷ 178.

In the accompanying motion of the vapor with a pressure $p$ = 0,196 ÷ 0,245 MPa and a current of liquid flowing from the cylindrical nozzle from top to bottom:

$$4 \cdot St = 0{,}33 \cdot \left( \frac{l}{d_c} \right)^{-0{,}59} \cdot \mathrm{Re}^{-0{,}17} \cdot \mathrm{Pr}^{0{,}09} \cdot K^{0{,}13} \cdot We^{0{,}33}, \qquad (21)$$

at $We$ = 2.7 ÷ 7.4; $\frac{l}{d_c}$ = 4.5 ÷ 120.0; $d_c$ = 2.18;4;6 mm ; $l \geq 50$ d; Re > $10^4$.

The publications [35; 36] present the data of studies of heat exchange under condensation on laminar [35] and turbulent [36] jets of liquid, taking into account the initial inlet section:

$$St = f_1 \cdot f_2 \cdot f_3 \cdot \left( \frac{x}{2 \cdot R_0} \right)^{-0{,}8}, \qquad (22)$$

where $f_1, f_2, f_3,$ are the functions of the numbers Re, Pr, $We$ and the complex $\frac{x}{2R_0}$, obtained by changing $We$ from $10^{-3}$ to 5; Re from 200 to 1000; Pr from 1 to 50.

In [37] the formula is obtained:

$$St = 0{,}047 \cdot \left( \frac{1}{2 \cdot R_0} \right)^{-0{,}52} \cdot \mathrm{Re}^{-0{,}033} \cdot \mathrm{Pr}^{-0{,}074} \cdot Fr^{-0{,}064}. \qquad (23)$$

The calculation of formulas (22) and (23) was consistent with previous experiments by Kutateladze S. S., Isachenko V. P., Dementyeva K. V.

Recent studies [38] have been associated with the development of a mathematical model for the calculation of jet condensation and the analysis of past developments with its application.

As is known, there is no study of heat transfer during condensation of a dispersed steam stream on a swirling stream of water. With regard to the above dependencies, we can conclude about their individuality due, firstly, to the method of obtaining, and secondly, to the differences in the method of determining the input parameters.

**Conclusions**. Formula (8) is obtained for a flat turbulent jet of liquid according to the statement that a turbulent mixing layer develops at the boundary. Formulas (9, 10, 15, 16) have the same theoretical origin under the same conditions at the boundary of the two phases. Formulas (11, 12) describe experimental studies of condensation under vacuum conditions.

The determination of the physical parameters of the liquid by temperature also differs:

- for formulas (10–14) it is the average liquid temperature $t_{avg} = (t_1 + t_0) / 2$;
- for formulas (9, 15-21) it is the liquid temperature at the inlet $t_0$;
- for formula (14) it is the saturation temperature $t_s$.

The quantities $\rho_n$, $\sigma$, $r$ were determined by the saturation temperature $t_s$ of the vapor.

## References:


1. Дульнев Г. Н. Принцип местного влияния в методе поэтапного моделирования / Г. Н. Дульнев, Е. В. Сахова, А.В. Сигалов // ИФЖ. — 1983. — Т. 45, № 6. — С. 831–836.
2. Таубман Е. И. Изв. АН СССР. Сер. Энергетика и транспорт / Е. И. Таубман. — 1965. — № 5. — С. 121–130.
3. Исаченко В. П. Теплопередача / В. П. Исаченко, В. А. Осипова, А. С. Сукомел. — М. : Энергия, 1975. — 481 с.
4. Кутателадзе С. С. Теплоотдача при конденсации и кипении / С. С. Кутателадзе. — М. ; Л. : Машгиз, 1952. — 232 с.
5. Исаченко В. П. Теплообмен при конденсации / В. П. Исаченко. — М. : Энергия, 1977. — 239 с.
6. Кутепов А. М. Гидродинамика и теплообмен при парообразовании / А. М. Кутепов, Л. С. Стерман, Н. Г. Стюшин. — М. : Высшая школа, 1977. — 352 с.
7. Вопросы физики кипения. — М. : Мир, 1964. — С. 443.
8. Григорьев В. А. Кипение криогенных жидкостей / В. А. Григорьев, Ю. М. Павлов, Е. В. Аметистов. — М. : Энергия, 1977. — 289 с.
9. Берман Л. Д. Испарительное охлаждение циркуляционной воды / Л. Д. Берман. — М. : Энергия, 1957. — 319 с.
10. Соснин Ю. П. Контактные водонагреватели / Ю. П. Соснин. — М. : Стройиздат, 1974. — 359 с.

11. Федоткин И. М. Теплогидродинамические процессы в выпарных аппаратах / И. М. Федоткин, С. И. Ткаченко. — Киев : Техніка, 1975. — 212 с.
12. Чернобыльский И. И. Выпарные установки / И. И. Чернобыльский. — Киев : Изд-во КГУ, 1960. — 260 с.
13. Черный Г. Г. ДАН СССР / Г. Г. Черный. — 1955. — Т. 101, № 1. — С. 39–42.
14. Иса И. Теплопередача / И. Иса, Цзинь-Жань Чжень. — 1972. — № 1. — С. 54–58.
15. Shekriladze I. G. Int. I. Heat and Mass Transfer / I. G. Shekriladze, V. I. Gomelauri. — 1966. — V. 9, № 6. — P. 581–591.
16. Бойко Л. Д. Теплообмен в элементах энергетических установок / Л. Д. Бойко. — М. : Наука, 1966. — С. 197–212.
17. Волков Д. И. Теплообмен при конденсации и кипении / Д. И. Волков // Труды ЦКТИ. — Л. : Изд. ОНТИ ЦКТИ, 1965. — Вып. 57. — С. 149–159.
18. Кутателадзе С. С. Вопросы теплоотдачи и гидравлики двухфазных сред / С. С. Кутателадзе. — М. ; Л. : Госэнергоиздат, 1966. — С. 138–156.
19. Исаченко В. П. Теплообмен и гидродинамика / В. П. Исаченко, С. А. Сотсков, Е. В. Якушева // Труды МЭИ. — М., 1975. — Вып. 235. — С. 145–152.
20. Труб И. А. Теплофизика и теплотехника / И. А. Труб. Киев : Наукова думка, 1964. — С. 33–39.
21. Труб И. А. Каскадные конденсаторы смешения / И. А. Труб. — М. : Пищевая промышленность.
22. Толубинский В. И. Теплообмен при кипении / В. И. Толубинский. — Киев : Наукова думка, 1980. — 315 с.
23. Ильин И.Н., Турлайс Д.П., Ротбаум Я.М. Кипение и конденсация / И. Н. Ильин, Д. П. Турлайс, Я. М. Ротбаум. — Рига, 1979. — С. 60–67.
24. Тананайко Ю.М. Методы расчета и исследования пленочных процессов / Ю. М. Тананайко, Е. Г. Воронцов. — Киев : Техніка, 1975. — 311 с.
25. Федоткин И.М., Фирисюк В.Р. Тепло-массообмен / И. М. Федоткин, В. Р. Фирисюк. — Киев : Наукова думка, 1968. — С. 181–191.
26. Чжунь Ч. Труды американского общества инж.-механиков, Сер. С. Теплопередача / Ч. Чжунь, Р. Себан. — 1972. — № 4. — С. 106–110.
27. Чанг П. ЭИ. Сер. Теплоэнергетика / П. Чанг. — 1963. — Вып. 9. — С. 1–20.
28. Агафонов В. А. Судовые конденсационные установки / В. А. Агафонов, В. Г. Ермилов, Е. В. Панков. — Л. : Судпромгиз, 1963. — 490 с.
29. Таубман Е.И. Тепло- и массоперенос / Е. И. Таубман. — Минск, 1966. — Т. 6. — С. 306–315.
30. Рабинович Г. Д. ЖТФ / Г. Д. Рабинович. — 1953. — Т. 23. — Вып. 3. — С. 541–549.
31. Шумская Л. С. Вопросы теплоотдачи и гидравлики двухфазных сред / Л. С. Шумская. — М. : Госэнергоиздат, 1961. — С. 350–392.
32. Коченов И. С. Доклады АН СССР / И. С. Коченов. — 1956. — Т. 107. — № 5. — С. 689–692.
33. Стренч П. Вопросы физики кипения / П. Стренч, А. Орел, Дж. Аэстуотер. — М. : Мир, 1964. — С. 331–353.
34. Фритц В. Вопросы физики кипения / В. Фритц, В. Энде. — М. : Мир, 1964. — С. 162–188.
35. Флоршюц Л. Труды американского общества инж.-механиков. Сер. С. Теплопередача / Л. Флоршюц, Б. Чао. — 1965. — Т. 87. — № 2. — С. 58–72.

36. Ладиев Р. Я. Теплообмен и гидродинамика в двухфазных средах / Р. Я. Ладиев. — Киев : Наукова думка, 1967. — С. 63–72.
37. Гаусбрандт Э. Выпаривание, конденсация и охлаждение / Э. Гаусбрандт, М. Гирш. — М. : ОНТИ Главн. ред. хим. лит-ры, 1936. — 380 с.
38. Зингер Н. И. Вопросы теплообмена при изменении агрегатного состояния вещества / Н. И. Зингер. — М. ; Л. : Госэнергоиздат, 1963. — С. 81–91.

## КОНДЕНСАЦІЙНИЙ ТЕПЛОМАСООБМІН ПРИ ПРЯМОМУ КОНТАКТІ РЕАГУЮЧИХ ФАЗ

Конденсацію при прямому контакті фаз умовно можна розділити на такі види: конденсація струменю пари в об'ємі недогрітої рідини; конденсація бульбашок пари в рідині; конденсація пари на краплях рідини (диспергована рідина); конденсація пари на струмені рідини.

При візуальних експериментах дослідження процесу конденсації струменя пари у просторі відмічалася наявність білої емульсії при зіткненні пари з рідиною, за рахунок дроблення струменю пари на маленькі бульбашки. Висока інтенсивність процесу теплообміну пояснювалась різким збільшенням поверхні контакту. Якщо розглядати структуру потоку з урахуванням двофазної області, можна відмітити наявність як гладкої конічної поверхні розділу між фазами, так і утворення дисперсних бульбашок та крапель в потоці. Це дозволяє визначити залежність геометрії зони контакту фаз від температурного напору.

Збільшення площі поверхні контакту фаз може бути досягнуто внаслідок диспергування однієї з контактуючих фаз. Існуючі апарати з розпиленням рідини мають значні енерговитрати як наслідок виконання деякої роботи для подолання поверхневого натягу, який примушує рідину зменшувати вільну поверхню. Так тепломасообмін між краплиною рідини і насиченою парою визначається розподілом теплоти вздовж радіуса краплі. Пара конденсується на поверхні краплі рідини, при цьому теплота конденсація, що вивільнюється, має відводитись всередину краплі. Згідно з рівнянням теплопровідності при відповідних умовах розглянутої задачі, інтенсивність конденсації визначається швидкість стоку теплоти у краплю. Дослідження теплообміну на диспергованих струменях рідини довели високу інтенсивність процесу.

Конденсація на струмені рідини використовується в багатьох промислових апаратах (деаератори, конденсатори змішувального типу, струменеві нагрівачі).

Теоретичні та експериментальні дослідження цього виду конденсації небагаточисельні. Дослідження теплообміну при конденсації диспергованого струменя пари на закрученому струмені води взагалі відсутні.

Результати дослідів теплообміну при контактній конденсації пари на струменях води, що складається з суцільної ділянки та ділянки, що розпадається на краплини, представляються критеріальним рівнянням. Останні дослідження пов'язані з розробкою математичної моделі розрахунку струминної конденсації та аналізу минулих розробок з її застосуванням.

**Ключові слова:** *конденсація, пара, струмінь, рідина, контакт фаз, теплообмін.*

Отримано: 20.08.2019