ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
Vol. 9, Issue 11 , November 2022

Methodology for conduction and calculation of the results of an experimental study for condensation of vapors on vertical tubes with specially profiled ribs

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ABSTRACT: The article considers the issues of the methodology for conducting and calculating the results of an experimental study on the condensation of water vapor on vertical tubes with specially profiled ribs of a heat exchanger. Based on the analysis of works on the study of heat transfer during laminar condensation of water vapor in the form of a laminar - flowing liquid film both inside and on the outer surfaces of vertical tubes with a stationary steam flow, a laboratory installation was created, on which experimental studies were carried out. One of the methods of intensifying the heat exchange process is to improve the geometric configuration of the heat exchange surface on the heating side of the heat exchange tube, where steam condensation occurs due to heat transfer, based on reducing the thermal resistance of the wall layers of the formed condensate. This method is based on increasing the heat exchange surface area of the heat exchanger tubes with specially profiled ribs (ribbing). At the same time, an important scientific problem is solved - the disruption of the flow of steam laminarily drained from the heat exchange surface of the condensate, which leads to the contact of steam directly with the cold surface of the tube and increases heat transfer. The article describes the method of conducting experimental studies, provides a method for processing the results of experiments, as well as calculated data and graphical dependencies that display the obtained data.

KEY WORDS: heat exchangers, heat transfer, condensation, condensate, ribs, steam, caps, specially profiled ribs.

I. INTRODUCTION

The intensification of heat transfer processes leads to a decrease in the size and weight of heat exchangers with an increase in their thermal power, which is a very relevant issue.
In many industries, the issue of heat transfer from one source to another is acute. At the same time, one of the problems that need to be solved is to increase the efficiency of heat exchange. In existing heat exchange installations with vertical tubes, the flow of condensate formed on the cooling surface of the tube and its thickness strongly affect the efficiency of heat exchange. The thickness of the formed condensate film significantly increases the thermal resistance and reduces heat exchange. The organization of the discontinuity of the condensate flow is one of the main goals of the study.
This issue is considered in many published works, where various methods and ways are proposed that lead to an improvement in heat transfer during condensation of heating steam in tubular devices.
It is known that with abundant condensation of heating steam, condensate forms a liquid film on the surface of a vertical tube and, as it flows downward, the thickness of the condensate layer increases, which increases thermal resistance, which affects the heat transfer process [1 - 7].
Studied the heat transfer performance of pure water condensation outside integral ribbed tube and pin fin tube. They pointed out that pin rib tube has less condensate retention compared integral rib tube with identical other parameter [8 - 16].
Heat exchangers are usually part of process systems, and tubular heat exchangers are one of the most common types in industry. The most efficient tubular heat exchangers use the latent heat of a liquid or gas (evaporation), for example, during the transition from a liquid phase to a gas phase (evaporation) or from a gas phase to a liquid phase

ISSN: 2350-0328
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(condensation). The thermal design of a tubular heat exchanger based on heat condensation requires knowledge of the phase transition process in tubes. This work is devoted to the experimental analysis of steam condensation in a vertical smooth and specially profiled small-diameter tube.
Objective. Development and research of highly efficient heat transfer surfaces.
When developing heat exchange equipment, especially heat exchangers with vertical tubes, heat exchange indicators in the process of condensation of water vapor on the surface of a vertical tube are important [17, 18, 19].
In the study of steam condensation on vertical tubes, a film flow is considered and the main object of research is single vertical tubes with specially profiled ribs, which were developed in order to intensify the heat transfer process.
The authors [20, 21] proposed an improvement in the design of the heat exchanger edge in the form of a truncated cone installed at a certain distance over the entire height of the tube (Figure 1). Laboratory installation [22] was created with a developed heat exchange tube on which tests of this tube were tested. The methodology for experimental studies is described in [23]. The data obtained in experiments were mathematically processed. As a result, the study of this type is finishing the casting in Figure 2 and 3.
Experimental studies were carried out using the setup described in [22, 23]. In order to further compare the results, the experiments were carried out on two types of tubes, with and without ribbing. Before the start of the experiments, the stand was adjusted, the measuring equipment and the selected method of processing the results were checked and tested. Comparison of the data on steam condensation on smooth tubes and tubes with specially profiled ribs will show the correctness of the chosen method of conducting experimental studies and processing the data obtained, excluding the effect on the results obtained.
The range of variation of the parameters of steam and cooling water for a tube with specially profiled ribs and a smooth tube in each individual experiment was set to be the same.

II. EXPERIMENTAL TECHNIQUE

The experiments were carried out on smooth tubes with a change in the heat flux density from 19.5 to 45 kW/m2 and a temperature difference during condensation of 30÷410C, and in experiments with tubes with specially profiled ribs, these ranges were 15÷30 kW/m2 and 30÷55 0C, respectively.The experiments were carried out with carefully established initial parameters of the heat carriers (flow rate, temperature, pressure).
Each of the experiments was carried out for 30-40 minutes. The measurements were carried out only after the stand and instruments were completely warmed up in a stationary mode. An intermediate interval of 15-20 minutes was maintained between the experiments. During the experiment, readings were taken from all measuring devices. Table 1 shows data for the heater power Q=500 W.

Table 1. Experimental research results

Parameters	Value							
Cooling water quantity, *kg/sec*	0.01133		0.0143		0.01667		0.0175	
Tube type	smooth	ribs	smooth	ribs	smooth	ribs	smooth	ribs
Inlet temperature of water vapor, ^{0}C	93		93		93		93	
Condensate outlet temperature, ^{0}C	45	48	45.5	49.3	46	50	47	51.2
Cooling water inlet and outlet temperature, ^{0}C	27/34	27/36	27/33	27/34	27/32.3	27/33.2	27/32	27/33.5
The amount of condensate, *kg/h*	0.456	0.483	0.463	0.499	0.477	0.525	0.49	0.532
The heat transfer coefficient of water vapor to the wall α_v, $W/(m^2 \cdot K)$	701	764	716	773	732	782	770	745
The heat transfer coefficient from wall to water α_w, $W/(m^2 \cdot K)$	1165	1166	1375	1377	1500	1501	1543	1547
The overall heat transfer coefficient k, $W/(m^2 \cdot K)$	437	613.4	470	671	492	704	514	717

Figure 2 shows the calculated values of the overall heat transfer coefficient for smooth, with ring-shaped caps and tubes with specially profiled ribs, obtained on the basis of experimental data carried out at Q = 500 W and a cooling water flow rate Gw = 0.01133 ÷ 0.0175 kg / sec.

ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
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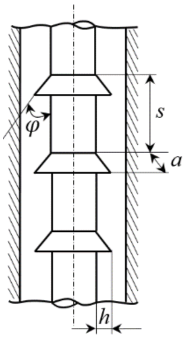

Fig. 1. Tubes with ring-shaped caps

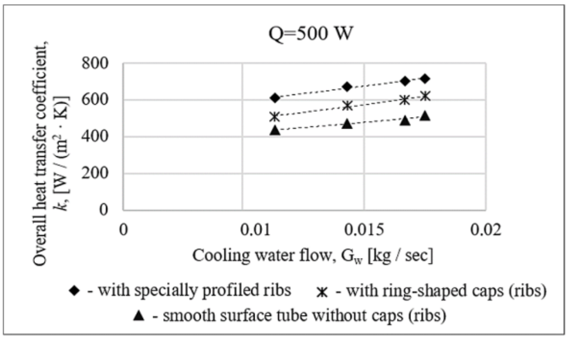

Fig. 2. Comparison of the overall heat transfer coefficient for smooth, with ring-shaped caps and tubes with specially profiled ribs

Figure 3 shows the experimental data obtained at a heater power of 500 W. In these experiments, the temperature of the cooling water remained constant and the flow rate varied. With an increase in the flow rate of cooling water, the heat transfer coefficient increases, both for a smooth tube, with ring-shaped caps and for a tube with specially profiled ribs.

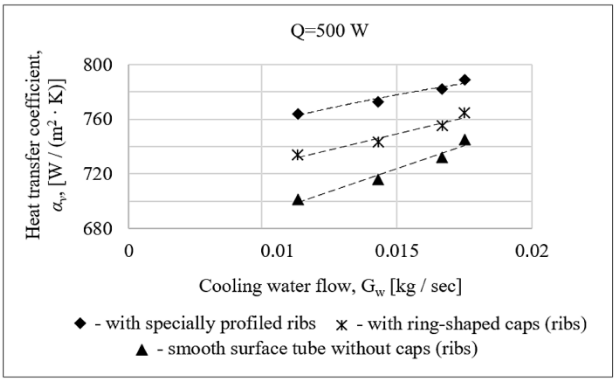

Fig. 3. Graph of the change in the heat transfer coefficient of water vapor in a smooth, with ring-shaped caps and a tube with specially profiled ribs with a change in the cooling water flow rate

The experimental data were processed according to the following procedure:

III. SOLUTION METHODS

If one of the heat carriers changes the phase state, for example, vapor condensation occurs when it is cooled with water, we have

$$Q_v = G_v(h_v - h_c) = G_w c_w (t_{out} - t_{in}) \tag{1}$$

where G_v, G_w - steam and cooling water consumption, *kg/sec*; h_v, h_c - enthalpy of steam at the entrance to the heat exchanger and condensate at the exit from it, *kJ/kg*; c_w - heat capacity of cooling water, $kJ/(kg \cdot {}^0C)$; t_{in}, t_{out} - temperature of cooling water at the inlet and outlet of the heat exchanger, ${}^0C$.

When determining the heat balance of the cooling water entering the heat exchanger, the temperatures of the entering and leaving cooling water were measured using a DS18B20 temperature sensor.

$$Q_w = G_w c_w (t_{out} - t_{in}) \tag{2}$$

The assessment of the convergence of the heat balance for water and steam was carried out using the heat balance discrepancy, determined by the formula [24]:

$$\delta = \frac{2|Q_v - Q_w|}{Q_v + Q_w} \cdot 100 \tag{3}$$

The measurement accuracy was considered acceptable when the heat balance discrepancy did not exceed 5%. The value of the heat transfer coefficient during condensation α_v [W / (m^2 · 0C] can be predicted using equations obtained

ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
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theoretically or experimentally. The theoretical equation of the heat transfer coefficient during condensation of water vapor was obtained by Nusselt [25]. The Nusselt equation (formula (3)) is expressed by the quantity condensate and thermal resistance of laminar film condensate on the wall surface.

$$\alpha_v = 0.9428 \cdot \left[\frac{g \cdot \rho_c \cdot r \cdot \lambda_c^3}{\nu_c \cdot (t_v - t_{wal})h} \right]^{0.25} \quad (4)$$

An analysis of the Nusselt equation using formula (4) showed that it is valid for non-moving steam, since the steam flow in the tube causes waves on the condensate surface. The wave effect increases the heat transfer during condensation by about 20.6%, as shown by Milan Kubin [26] by the following formula:

$$\alpha_v = 1.137 \cdot \left[\frac{g \cdot \rho_c \cdot r \cdot \lambda_c^3}{\nu_c \cdot (t_v - t_{wal})h} \right]^{0.25} \quad (5)$$

Further, equation (6) theoretically determines the heat transfer coefficient during condensation, taking into account the wave effect. This equation is often used in engineering problems. The equation published by Hobler (6) [27] is valid for many types of liquids with a pressure of 0.07< p_v [MPa]<17 and a specific heat flux 1.0< q_v [kW/m^2] < 1000.

$$\alpha_v = 0.00252 \cdot \left(\frac{\rho_v \cdot r}{\rho_c - \rho_v} \cdot \frac{\rho_c}{\delta_c}\right)^{0.33} \cdot \frac{\lambda_c^{0.8} \cdot q_v^{0.7}}{\mu_c^{0.5} \cdot c_c^{0.167} \cdot t_v^{0.37}} \cdot p_v^{\frac{10}{t_v}} \quad (6)$$

The function $\alpha = C \cdot q^v$ is based on the specific heat flux *q [W / m²]*, and the constant of the prefix C depends on the type of surface and properties of the liquid, and is taken in many cases *C = 1.537* [28, 29].

The average heat transfer coefficient during steam condensation on the outer surface of vertical tubes with fine finning is determined by the formula [30, 31].

$$Nu_v = 0.34 \frac{a^{0.15} \cdot h^{1.1} \cdot \theta^{-0.667}}{H^{0.25} \cdot S \cdot \cos\varphi} \cdot We^{0.21} \cdot (Ga \cdot \Pr \cdot K)^{0.37} \quad (7)$$

where: $\bar{\theta} = 0.7 n^{-0.4} We^{-0.1}$ $\beta < 1, nWe^{0.25} \geq 1$; $\bar{\theta} = 0.7 \beta^{-0.07} (nWe^{0.25})^m$ $\beta \geq 1, nWe^{0.25} \geq 1$ $m = -0.4\beta^{-0.15}$;

$$\bar{\theta} = 1 - 0.23\beta^{-0.36} (nWe^{0.25})^{1.2} \quad \beta \geq 1, nWe^{0.25} < 1;$$

$$Ga = \frac{g d_{эк}^3}{\nu'^2}; \qquad n = \left[\frac{\rho'^2 \cdot g \cdot r \cdot \lambda'^3 \cdot h^7 \cdot \cos\varphi}{4\mu' b^4 \lambda_{wal}^4 \Delta t_0} \right]^{0.25};$$

$$We = \frac{\sigma \cdot \cos\varphi}{g \rho' b h (1 + \tan\varphi)}; \qquad \beta = \frac{h \cdot \tan\varphi}{b}; \; K = \frac{r}{c_v \Delta t_0};$$

where: *a* – half the width of the intercostal groove, *m*; *b* – half the thickness of the edge at the end, *m*; *h* - rib height, *m*; *H* – tube length, *m*; *S* – rib spacing, *m*; φ – acute angle between the lateral surface of the rib and its axial plane; ρ' - saturation line density, *kg/m³*; ρ'' - dry saturated steam density, *kg/m³*; λ_T - metal thermal conductivity coefficient, W/(m^2·K); λ - thermal conductivity on the saturation line, W/(m^2·K); *r* – heat of vaporization, *J/kg*; *We* – Weber's criterion; Δt_0 – temperature head at the base of the rib, ^0C; σ – surface tension coefficient, N/m; *K* – phase transition criterion; μ - dynamic viscosity coefficient at the saturation line, *Pa·s*; ν' - kinematic viscosity coefficient at saturation line, m^2/s; *Ga* – Galileo criterion; c_p – heat capacity of steam, *J/(kg·K)*; *Pr* – Prandtl criterion.

Subscripts: *v* – vapour; *c* – condensate; *w* – water; *in* – inlet; *out* – outlet; *wal* – wall of tube.

In all experiments, a turbulent flow of cooling water was maintained ($Re > 10^4$), therefore, to calculate the heat transfer coefficient from the water side, the formula of M.A. Mikheev [32].

$$\alpha_w = 0.021 \mathrm{Re}_w^{0.8} \Pr_w^{0.43} \left(\frac{\Pr_w}{\Pr_{wal}}\right)^{0.25} \frac{\lambda_w}{d_{in}} \quad (8)$$

The temperature difference between the inlet and outlet of the cooling water did not exceed $10\ ^0C$, therefore, to determine the average temperature difference between the heat exchanging media, the following formula was used:

$$\Delta \bar{t} = \frac{(t_v - t_{out}) - (t_c - t_{in})}{\ln\left(\frac{t_v - t_{out}}{t_c - t_{in}}\right)} \quad (9)$$

ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
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In a heat exchanger installed on an experimental stand, the condition (d_{out} / d_{in}) <2 is fulfilled for the inner tube, therefore, the heat transfer coefficient is calculated using the following formula as heat transfer through a flat wall, taking into account the ribbing coefficient with specially profiled finned tubes:

$$k = \frac{1}{\dfrac{1}{\alpha_v \dfrac{f_{ribs}}{f_{smooth}}} + \dfrac{\delta}{\lambda_{metal}} + \dfrac{1}{\alpha_w} + R_{resis}} \qquad (10)$$

where f_{ribs} / f_{smooth} - is the coefficient of ribbing taking into account the increase in the heat exchange area; f_{ribs} - is the area with ribbing and f_{smooth} - is the area of a smooth tube; R_{resis} – thermal pollution resistance, $(m^2 \cdot K)/W$.

IV. CONCLUSION

The use of specially profiled ribs makes it possible to increase the heat exchange surface, increase the contact of steam with the cold cooling surface, due to the discontinuity of the flow of the flowing film, which creates an area free of condensate under the rib, where intense condensation occurs, which contributes to an increase in the efficiency of heat exchange with an increase in condensate output and heat transfer coefficient compared to smooth tubes. The proposed method for increasing the efficiency of heat exchange can be applied to various heat exchangers with vertical cooling tubes.

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