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Study of Steam Condensation on Vertical Finned Tubes

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ABSTRACT

The article is devoted to the methodology of conducting and processing the results of an experimental study of the process of condensation of water vapour on vertical pipes with specially profiled fins of a heat exchanger. Based on the analysis of heat transfer during laminar condensation of water vapour in the form of a layer of flowing liquid both inside and on the outer surface of vertical pipes with a stationary steam flow, a laboratory installation was developed on which experimental studies were carried out. One of the ways to intensify heat transfer is to optimize the geometry of the heat exchange surface on the condensation side, which reduces the thermal resistance of the wall layers of the resulting condensate. This method is based on increasing the heat exchange area by using specially shaped fins on the surface of the heat exchanger tubes. As a result, an important scientific problem is being solved – disruption of the continuous flow of laminar condensate, which contributes to the direct contact of steam with the cooled surface of the pipe and increases heat transfer. The article describes the methodology of conducting experiments, describes the methods of processing the results obtained, as well as provides calculated data and graphical dependencies illustrating the experimental results.

Keywords: heat exchangers, heat transfer, condensation, condensate, fins, steam.

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Introduction

One of the key challenges in heat transfer intensification is the simultaneous increase in thermal power, which contributes to a reduction in both the size and weight of heat exchangers. Numerous studies have proposed various methods to enhance heat transfer efficiency during the condensation of heating steam in tubular devices.

During the condensation of heating steam inside a vertical tube, a liquid condensate film forms, progressively increasing in thickness as it moves downward. This alters the thermal resistance, consequently affecting the overall heat transfer process [[1], [2], [3]].

Research on the condensation performance of pure water outside integral-fin and pin-fin tubes has shown that, under identical conditions, pin-fin tubes retain less condensate than integral-fin tubes, leading to improved heat transfer efficiency [[4], [5], [6]].

Heat exchangers are fundamental components of process systems, with tubular heat exchangers being among the most commonly used in industrial applications. The most efficient designs leverage phase transitions, such as evaporation and condensation, to optimize heat transfer. The thermal design of tubular heat exchangers, particularly those operating on condensation principles, requires a thorough understanding of phase transition processes within tubes. This study focuses on the experimental analysis of steam condensation in both smooth and specially profiled small-diameter vertical tubes [7].

The development and optimization of high-performance heat transfer surfaces play a crucial role in improving heat exchange equipment, particularly heat exchangers with vertical tubes. Heat transfer characteristics during the condensation of water vapour on vertical tube surfaces are of particular significance [[8], [9]].

In studies of steam condensation on vertical tubes, the primary research focus is on film flow dynamics. Single vertical tubes with specially profiled fins have been developed to enhance heat transfer efficiency. Experimental investigations were conducted using the setup described in [[10], [11], [12], [13]]. To ensure the reliability of results, tests were performed on both finned and smooth tubes. Before experimentation, the setup was calibrated, measuring instruments were tested, and data processing methodologies were verified. Comparing condensation performance between smooth tubes and tubes with specially profiled fins helps validate the experimental approach while minimizing external influences on the results.

For each experiment, the steam and cooling water parameters were maintained at consistent levels for both smooth and finned tubes, ensuring accurate comparisons of their heat transfer performance.

The experimental part

In the conducted research, the investigated parameters varied within the following ranges: water mass flow rate G_{water} = 0,01133 to 0,025 kg/s, steam mass flow rate D_{steam} = 0,0000833 ÷ 0,000115 kg/s, and pressure P = 0,05 to 0,13 MPa. The velocity of the cooling liquid was in the range of W_{water} = 0,14 to 0,32 m/s, while the Reynolds number for cooling water varied from Re_{water} = 1500 to 3300. The steam velocity was measured within W_{steam} = 0,07 to 0,09 m/s, with the corresponding Reynolds number ranging between Re_{steam} = 125 and 170, The steam temperature was recorded between t_{steam} = 94 and 105 ^{0}C.

The geometric parameters of an edge with an improved surface (EIS) are assumed in Table 1 [14].

Boundary conditions were established for different tubes used in the experiments. The velocity of steam entering the experimental flask was within the range of Re_{steam} = 0,07 to 0,09 m/s corresponding to Reynolds numbers between W_{steam} = 125 and 170. The power output of the steam generator was increased to Q = 300 to 630 W.

When a heat carrier undergoes a phase transition, such as steam condensation due to cooling with water, the heat transfer process is described by the equation:

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

where G_v, G_w - mass flow rates of steam and cooling water, kg/sec; h_v, h_c - enthalpy of steam entering the heat exchanger and condensate exiting, kJ/kg; c_w - specific heat capacity of cooling water, $kJ/(kg^0C)$; t_{in}, t_{out} - temperatures of cooling water at the inlet and outlet of the heat exchanger, ^0C.

To determine the heat balance, the temperatures of the cooling water entering and exiting the heat exchanger were measured using a DS18B20 temperature sensor. The heat balance equation for water is given by:

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

The heat balance accuracy between water and steam was assessed using the discrepancy formula [15]:

$$\bar{\delta} = \frac{2|Q_v - Q_w|}{Q_v + Q_w} \cdot 100\ \%$$

Table 1 - Geometric Parameters of a Vertical Tube with an Enhanced Surface (EIS)

Geometric parameters	**A**	**B**	**C**	**D**	**E**	**F**	**G**
The outer diameter of the EIS d_{EIS}, mm	18 – 22	18 – 22	18 – 22	18 – 22	18 – 22	18 – 22	18 – 22
Diameter of smooth tube d_{out}/d_{in}, mm	12/10	12/10	12/10	12/10	12/10	12/10	12/10
The radius of curvature of the edge end r, mm	3 – 4	3 – 4	3 – 4	3 – 4	3 – 4	3 – 4	3 – 4
The angle between the smooth tube and the fin φ, degree	30 - 35	30 - 35	30 - 35	30 - 35	30 - 35	30 - 35	30 - 35
Distance between fins S_2, mm	150	100	50	43	25	20	16.6
Fin sheet thickness δ, мм	0.001	0,001	0.001	0,001	0.001	0.001	0.001
Tube Length, ℓ, mm	300	300	300	300	300	300	300
Forming height of the EIS, α, mm	5-7	5-7	5-7	5-7	5-7	5-7	5-7
EIS surface area, $f_{EIS} = 10^{-4}$, m^2	4.396	8.792	21.98	28.4	48.4	61.54	79.9
Distance between the base of the generatrix and the smooth tube, b, mm.	3 - 5	3 - 5	3 - 5	3 - 5	3 - 5	3 - 5	3 - 5
Smooth tube surface area, $f_0 = 10^{-4}$, $м^2$	1.13	1.13	1.13	1.13	1.13	1.13	1.13
The total surface area of the vertical tube with EIS, $f = f_{EIS} + f_0$, m^2	0.0117	0.0121	0.0139	0.015	0.0164	0.0181	0.02
Finning ratio f / f_0	1.04	1.07	1.23	1.34	1.45	1.6	1.76

The measurement error was considered acceptable if the heat balance discrepancy did not exceed 5%.

The heat transfer coefficient for condensation, α_v, measured in [Wt / (m^2 · ^0C] can be predicted theoretically or experimentally. The classical theoretical model for condensation heat transfer was proposed [16], expressed as:

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

However, analysis of this formula equation (4) indicates that it applies primarily to stationary steam. The presence of vapour flow induces wave formation on the condensate surface, which enhances heat transfer by approximately 20.6%. The authors [17] proposed the following modified equation to account for this effect:

$$\alpha_v = 1.137 \cdot \left[\frac{g \cdot \rho_c \cdot r \cdot \lambda_c^3}{v_c \cdot (t_v - t_{wal}) h}\right]^{0.25} \quad (4)$$

Further refinements led to the development of a theoretical equation for the heat transfer coefficient, incorporating wave effects. Hobler's equation [17] (Equation 5) is widely applied in engineering calculations and is valid for various liquids under pressure conditions $0.07 < p_v$ [MPa] < 17 and specific heat flux values $1.0 < q_v$ [kWt / m^2] < 1000.

$$б_v = 0.00252 \cdot \left(\frac{с_v \cdot r}{с_c - с_v} \cdot \frac{с_c}{д_c}\right)^{0.33} \cdot \frac{л_c^{0.8} \cdot q_v^{0.7}}{м_c^{0.5} \cdot c_c^{0.167} \cdot t_v^{0.37}} \cdot p_v^{\frac{10}{t_v}} \quad (5)$$

The heat transfer coefficient function follows the relation $= C \cdot q^n$, where the specific heat flux q_v, [kWt / m^2] is considered. The constant C depends on the surface type and liquid properties, often taken as $C = 1.537$.

For vertical tubes with fine fins, the average heat transfer coefficient during steam condensation is determined using the following equation [[18], [19]].

$$Nu_v = 0{,}34 \frac{a^{0{,}15} \cdot h^{1{,}1} \cdot \theta^{-0{,}667}}{H^{0{,}25} \cdot S \cdot \cos\varphi} \cdot We^{0{,}21} \cdot (Ga \cdot Pr \cdot K)^{0{,}37} \quad 6)$$

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

$1, nWe^{0.25} \geq 1$; $m = -0.4\beta^{-0.15}$; $\bar{\theta} = 1 - 0.23\beta^{-0.36}(nWe^{0.25})^{1.2}$ at $\beta \geq 1, nWe^{0.25} < 1$;

$$n = \left[\frac{\rho'^2 \cdot g \cdot r \cdot \lambda'^3 \cdot h^7 \cdot \cos\varphi}{4 \cdot \mu' \cdot b^4 \cdot \lambda_{wal}^4 \Delta t_0}\right]^{0,25};$$

$$We = \frac{\sigma \cdot \cos\varphi}{g \cdot \rho \prime \cdot b \cdot h \cdot (1 + \tan\varphi)}; \quad \beta = \frac{h \cdot \tan\varphi}{b}; \quad Ga = \frac{g d_{eq}^3}{\nu'^2}; K = \frac{r}{c_v \Delta t_0};$$

where a - half the width of the intercostal groove, *m*; b - half the thickness of the fin at the end, *m*; h - fin height, *m*; H - tube length, *m*; *s* - fin spacing, *m*; φ - acute angle between the fin's lateral surface and axial plane; ρ' - density along the saturation line, *kg/m³*; ρ'' - density of dry saturated steam, *kg/m³*; λ_{metal} - metal thermal conductivity, *Wt/(m·⁰C)*; λ - thermal conductivity along the saturation line, *Wt/(m·⁰C)*; *r* - heat of vaporization, *J/kg*; We - Weber number; Δt_0 - temperature difference at the fin base, *⁰C*; σ - surface tension, N/m; K - phase transition criterion; μ' - dynamic viscosity at the saturation line, *Pa·s*; ν' - kinematic viscosity at the saturation line, m²/s; Ga - Galileo number; c_v - specific heat capacity of steam, *J/(kg·⁰C)*; Pr - Prandtl number.

Subscripts: v - vapour; c - condensate; w - water; *eq* – equivalent; $aver$ – average; in - inlet; out - outlet; wal - tube wall.

In all conducted experiments, a turbulent flow of cooling water was maintained ($Re_{\text{в}} > 10^4$). Therefore, the heat transfer coefficient on the water side was determined using the following equation [20]:

$$\alpha_w = 0.021 Re_w^{0.8} Pr_w^{0.43} \left(\frac{Pr_w}{Pr_{wal}}\right)^{0.25} \frac{\lambda_w}{d_{in}}. \quad (7)$$

The temperature difference between the inlet and outlet cooling water remained within10 ⁰C. Consequently, the logarithmic mean temperature difference (LMTD) between the heat exchange media was calculated using the formula:

$$\Delta\bar{t} = \frac{(t_v - t_{out}) - (t_c - t_{in})}{ln\left(\frac{t_v - t_{out}}{t_c - t_{in}}\right)} \quad (8)$$

For the inner tube, the overall heat transfer coefficient was determined based on heat transfer through a flat wall, incorporating the effect of finning in specially profiled finned tubes:

$$k = \frac{1}{\frac{1}{\alpha_v \frac{f_{fins}}{f_{smooth}}} + \frac{\delta}{\lambda_{metal}} + \frac{1}{\alpha_w} + R_{ther}} \quad (9)$$

where $\frac{f_{fins}}{f_{smooth}}$- is the finning coefficient, accounting for the increased heat exchange area; f_{fins} - represents the finned area, while f_{smooth} corresponds to a smooth tube surface; R_{ther} - denotes thermal resistance due to fouling, expressed in, *(m² ·K)/Wt*.

Discussion of the results

Generalized results from experimental studies are presented in Figures 1 and 2. The empirical equation describing the relationship between thermal resistance R_{ther} and film thickness (δ) is given as:

$$R_{ther} = 14{,}7409 \ln(\delta) + 49{,}9661 \quad (10)$$

Additionally, the heat transfer coefficient α dependence on film thickness was determined empirically as:

$$\alpha = 731{,}1334\delta^{-0{,}95936} \quad (11)$$

The relative errors associated with equations (10) and (11) were 2.9% and 0.14%, respectively. The Fisher criterion was used to validate equation (10), yielding Fr = 5.3033 with a reliability probability of P = 0.95, while the tabulated value was Ft = 10.13. Similarly, equation (11) produced Fp= 5.1884 confirming its adequacy against the same tabulated value.

Experiments were conducted on a 300 mm-long test tube, where fin spacing (S_2) was optimized using the Nusselt criterion. The study revealed that maximum relative heat transfer coefficient values$\alpha/\alpha_0 = 1{,}6 \div 1{,}8$ corresponded to optimal fin spacing $S_2 = 50 \div 35$ mm. Alternatively, this range aligns with finning coefficient values of $f/f_0 = 1{,}23 \div 1{,}35$ as shown in Figure 2.

The finning coefficient (f/f₀) represents the ratio of the total finned surface area (f) to that of a smooth tube (f₀). The dependency of the relative heat transfer coefficient on the finning coefficient was determined through the least squares method, yielding the empirical equation:

$$\alpha/\alpha_0 = 8{,}1029(f/f_0)^3 - 37{,}42(f/f_0)^2 + +56{,}194(f/f_0) - 25{,}914 \quad (12)$$

A comparative analysis of experimental data and mathematical modelling results demonstrated their consistency. Adequacy was verified using the Fisher criterion, with an average relative error ranging between 2.03% and 3.8%.

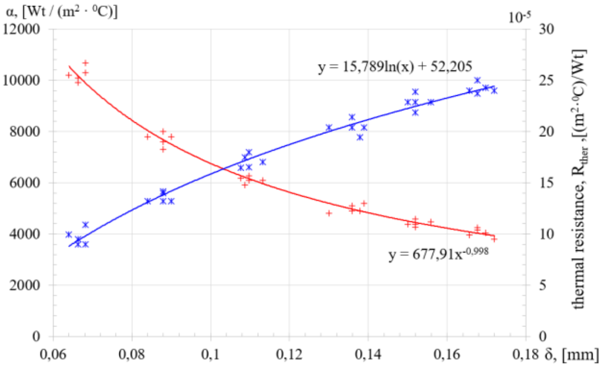

Figure 1 - Dependence of α heat transfer coefficient and R_{ther} thermal resistance on film thickness

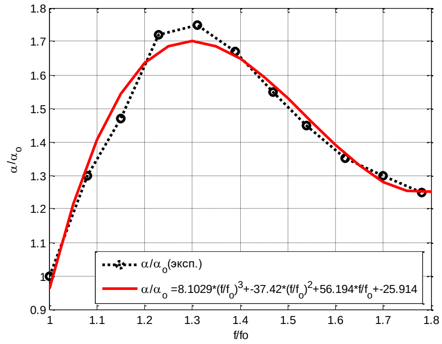

Figure 2 - Dependence of the relative change in the heat transfer coefficient on the finning coefficient

Conclusion

Based on the experimental results, it was determined that the average heat transfer coefficient for smooth tubes is 7258 $Wt/(m^2 \cdot ℃)$, while for vertical tubes with an enhanced finned surface, it reaches 8911 $Wt/(m^2 \cdot ℃)$. This indicates that the heat transfer efficiency of the improved surface is 23% higher compared to a smooth tube. Additionally, the condensate yield of a relatively smooth tube is 57% greater, and in comparison to surfaces shaped as truncated cones proposed by Mikheev and Mikheeva, it is 27% higher.

For vertical tubes, both theoretical and experimental data were used to determine and validate the optimal geometric parameters that enhance the efficiency of installations with finned surfaces. These include:

- The outer diameter of the rib relative to the perpendicular to the surface: (d_{EIS} =18÷20 mm);
- The rib inclination angle: (φ = 30 ÷ 35°);
- The finning coefficient: (f / f_0 =1,23÷1,35);
- The optimal vertical rib spacing: (S_2 = 35 ÷ 50 mm).

Conflict of interest. On behalf of all the authors, the correspondent author declares that there is no conflict of interest.

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ТҮЙІНДЕМЕ
Мақала жылу алмастырғыштың арнайы профильді қабырғалары бар тік құбырлардағы су буының конденсация процесіне эксперименттік зерттеу жүргізуге және оның нәтижелерін өңдеу әдістемесіне арналған. Стационарлық бу ағынында тік құбырлардың ішінде де, сыртқы бетінде де ағып жатқан сұйықтық қабаты түріндегі су буының ламинарлы конденсациясындағы жылу алмасуды талдау негізінде зертханалық қондырғы жасалды, онда эксперименттік зерттеулер жүргізілді. Жылу алмасуды күшейту әдістерінің бірі конденсация жағындағы жылу алмасу бетінің геометриясын оңтайландыру болып табылады, бұл алынған конденсаттың қабырға қабаттарының жылу кедергісін азайтуға

мүмкіндік береді. Бұл әдіс жылу алмастырғыш құбырларының бетіне арнайы профильді жиектерді қолдану арқылы жылу алмасу аймағын ұлғайтуға негізделген. Нәтижесінде маңызды ғылыми міндет шешіледі – ламинарлы ағып жатқан конденсаттың үздіксіз ағыны бұзылады, бұл будың құбырдың салқындатылған бетімен тікелей жанасуына және жылу беруді арттыруға ықпал етеді. Мақалада эксперименттерді жүргізу әдістемесі берілген, алынған нәтижелерді өңдеу әдістері сипатталған, сонымен қатар эксперименттік нәтижелерді бейнелейтін есептелген мәліметтер мен графикалық тәуелділіктер келтірілген.

Түйін сөздер: жылу алмастырғыштар, жылу беру, конденсация, конденсат, қатайту қабырғалар, бу.

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Изучение конденсации пара на вертикальных трубах с ребрами жесткости

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АННОТАЦИЯ

Статья посвящена методологии проведения и обработки результатов экспериментального исследования процесса конденсации водяного пара на вертикальных трубах с специально профилированными рёбрами теплообменника. На основе анализа теплообмена при ламинарной конденсации водяного пара в виде слоя стекающей жидкости как внутри, так и на внешней поверхности вертикальных труб при стационарном паровом потоке была разработана лабораторная установка, на которой проведены экспериментальные исследования. Одним из способов интенсификации теплообмена является оптимизация геометрии теплообменной поверхности на стороне конденсации, что позволяет снизить термическое сопротивление пристенных слоев образующегося конденсата. Этот метод основан на увеличении площади теплообмена за счёт применения специально профилированных рёбер на поверхности труб теплообменника. В результате решается важная научная задача – нарушение сплошного потока ламинарно стекающего конденсата, что способствует непосредственному контакту пара с охлажденной поверхностью трубы и увеличению теплопередачи. В статье изложена методика проведения экспериментов, описаны способы обработки полученных результатов, а также приведены расчётные данные и графические зависимости, иллюстрирующие экспериментальные результаты.

Ключевые слова: теплообменники, теплопередача, конденсация, конденсат, ребра жесткости, пар.

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