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HEAT EXCHANGE UNDER THE LONGITUDINAL MOVEMENT OF WET STEAM IN FINNING HEAT EXCHANGERS

The paper is devoted to the study on hydrodynamics and heat exchange of two-phase medium. While designing technological equipment, when the wet steam is used as the operating medium, the features of the interaction between liquid drops and the heat exchange surface are not considered in most cases. In full, this applies to steam turbines operating on the wet steam whose moisture content depends on the primary and secondary removal of liquid drops from the separation blocks.

Purpose. Improving the method of calculation of recuperative heat exchangers, if wet steam is used as the operating medium.

Methodology. It is based on the analysis of the physical model of moving the two-phase medium in the heat and mass exchange conditions, considering the design characteristics of the heat transfer surface.

Findings. The correlation of critical values of two-phase flow parameters was obtained to determine the lower boundary of the process of plucking the drops from the liquid film depending on the irrigation density, geometric characteristics of the channel and physical properties of the liquid and gas. Correlations were obtained for pipes with longitudinal finning as the Π-shaped profile, based on which we recommend optimizing the geometric characteristics of longitudinal finning.

Originality. Determining the limit modes of secondary removal formation during the movement of a two-phase medium in separation devices and the features of heat and mass transfer of wet steam in finning recuperative heat exchangers.

Practical value. The presented results make it possible to optimize the design of recuperative heat exchangers with longitudinal Π-shaped finning.

Keywords: *hydrodynamics of movement, two-phase flow, recuperative heat exchanger, wet steam*

Introduction. Developing the complex technological processes related to transferring and transforming the energy and mass requires an in-depth analysis of physical and chemical processes followed by developing the methodological foundations to create the equipment which is competitive by its characteristics. This mostly refers to technologies in which changing in the aggregate state and the thermophysical properties of substances is provided. Heat exchange processes in a wet steam, which are common enough at the industrial equipment, do not mostly consider the features of the processes between liquid drops and the heat exchange surface, which leads to an inconsistency between design parameters of the heat carrier and the actual values. This mostly applies to the steam turbines operating on the wet steam. To increase the efficiency and reliability of such type of turbine, a steam superheater separator (SSS) is installed between the high-pressure cylinders (HPC) and low-pressure cylinders (LPC). Analysing the processes occurring in separate elements of the SSS, it is possible to denote that the operating medium changes its parameters in the part of a wet steam in the first stage of the superheater, and the steam is superheated up to a value corresponding to the temperature and pressure at the input to the LPC only within the second stage. The operating parameters of the SSS and LPC slightly differ from design values in practice. The reasons for parameter deviations are mostly related to violating the efficiency of separate elements of the SSS. The SSS separation devices are designed to reduce steam humidity from initial values at the output of the HPC $(1 - x_{in})$ to the humidity values at the output of the SSS $(1 - x_{out}) < 1$ %. The initial steam humidity value for different types of turbines varies in the range $(1 - x_{in}) = 12–15$ %. Under reducing the load of the power units, the initial humidity hardly changes, and its value does not practically exceed the values which are typical for the nominal mode. Therefore, it is considered that the parameters of the nominal mode determine the geometric characteristics of the separation blocks. Appearing of the liquid drops in a steam is associated with maldistributions of distributing the wet steam by the separation blocks and the insufficient efficiency of the louver-type separators. The reasons for deteriorating the separation process are limitations related to the features of the louver design (length and height of the separation channel, distance between the plates, the angle by which the movement direction changes at the channel, the number of corrugations), the spectrum of distributing the drop size at the input, as well as a secondary removal of the drops from the surface of the film and the back edge of the louver plate. Then the wet steam enters the superheated part of the SSS. It consists of two sequentially located sections, where the steam from the extractions of the K-1000-60/3000 steam turbine is used as a heat carrier. Each section consists of 94 cassettes for the first stage and 93 cassettes for the second one of the SSS-220M, for the SSS-1000 it respectively consists of 80 cassettes for the first stage and 103 cassettes for the second one. The cassette is made of 37 tubes with longitudinal finning as a Π-shaped profile welded to the outer surface of the tube.

Analysis of process features at the first stage of the superheater separator. ***Heat and mass transfer of wet steam at the first stage of the superheater separator.*** The method for thermal calculation is based on joint solving the equations of the heat balance and heat transfer for the heat carrier (heating steam) and the operating medium (heated wet steam). The heat generated by the steam condensation on the vertical inner surface of the heat exchange pipe is transferred by forced convection up to the operating medium through the outer surface of the finned pipe. Structurally, the downward movement of the operating medium occurs in the first stage, and the upward movement occurs in the second one. It is assumed that the operating medium is superheated steam at the output from the second stage. The value of the amount of superheating in the second stage is $\Delta t_1 \approx 52°$

The heat amount that the operating medium will receive in the first stage is

$$Q_b = G_b \cdot (h_b'' - h_b'),$$

where h_b'' is enthalpy of steam at the output from the first stage, kJ/kg; h_b' is enthalpy of wet steam at the input to the first stage taking into account, kJ/kg; G_b is steam consumption, kg/s.

The standard method is proposed to determine the heat exchange area of the first and second stages, it is used for calculating the recuperative heat exchange devices with forced convection. The process of heat exchange occurs on the inner surface of the heat exchange pipe during the condensation of the steam coming from the turbine extractions, and the limit conditions of the second kind are, in fact, observed, while on the outside, the heat from the finned side is transferred to the wet steam by convection. Moreover, it is necessary to consider the influence of the design features of the finned pipe on the heat transfer process. According to the heat transfer equation, the amount of heat transferred in the heat transfer process is determined as follows

$$Q_b = k_1 \cdot \Delta t_1 \cdot F_1,$$

where k_1 is heat transfer coefficient, W/(m^2 · °C); Δt_1 is temperature drop, °C; F_1 is heat exchange surface area, m^2.

The temperature drop Δt_1 is calculated as the average logarithmic temperature drop at the known temperatures of the heat carrier at the input and at the output of the stage, the known temperature of the operating medium at the input and the accepted temperature at the output from the stage, which values are clarified by using the method of consecutive approximations.

For the convenience of calculations in the methods of determining the thermal capacity of heat exchangers with a cylindrical heat exchange surface, the linear heat transfer coefficient k_{l1} is used which is determined as

$$k_{l1} = \left(\frac{1}{\alpha_{1red} \cdot \varphi \cdot d_1} + \frac{1}{2 \cdot \lambda_m} \cdot \ln \frac{d_1}{d_2} + \frac{1}{\alpha_2 \cdot d_2} \right)^{-1}, \qquad (1)$$

where d_1 is the outer diameter of the tubular heating surface with finning, m; d_2 is inner diameter of the tubular heating surface, m; α_{1red} is the reduced coefficient of heat transfer from the side of the working body, W/(m^2 · °C); α_2 is the coefficient of heat transfer from the side of the heat carrier, W/(m^2 · °C); φ is the finning coefficient; λ_m is thermal conductivity coefficient, W/(m^2 · °C).

The appearance of overflows in the steam turbine condenser leads to a disturbing the water-chemical regime of the heat carrier of the second circuit of the VVER-1000 power unit and, as a result, it leads to the appearance of a layer of deposits of the hardness salts on the heat-mass exchange surfaces. Furthermore, vibration processes directly affect the strength of the connections of the longitudinal П-shaped fin with the outer surface of the heat exchange pipe. When performing verification calculations of the steam superheater separators, the general form of dependence (1) for determining the coefficient k_{l1} does not consider the influence of operational factors on the intensity of heat transfer processes. Considering the above, the calculation of the linear coefficient of the heat transfer for the SSS cassette must be performed by taking into account the above factors

$$k_{l1} = \left(\frac{1}{\alpha_{1red} \cdot \varphi \cdot d_1} + \frac{R_1}{d_1} + \frac{1}{2 \cdot \lambda_m} \cdot \ln \frac{d_1}{d_2} + \frac{R_2}{d_1} + \frac{R_3}{d_2} + \frac{1}{\alpha_2 \cdot d_2} \right)^{-1}, \qquad (2)$$

where R_1, R_3 are thermal resistance of deposits on the outer and inner surface of the pipe, (m^2 · °C); R_2 is thermal resistance that occurs when the contact of the finning welding joint with the outer surface of the pipe is broken, (m^2 · °C).

Determining the value of the heat transfer coefficient α_2 during the condensation of motionless steam is performed according to the standard method. However, on the outer surface, the heat exchange process is complicated by the presence of liquid drops in the two-phase flow. To determine the coefficient of heat transferring from the outer surface of the finned pipe to the wet steam α_1, it is necessary to consider its humidity at the input to the cassette $(1 - x_{out})$ and the drop size. The importance of evaluating the above factors lies in the fact that they directly affect the heat transfer process. To determine the intensity of heat exchange during the movement of the operating medium along the surface, [1] it is proposed to use the correlation for a homogeneous medium

$$\mathrm{Nu} = 0.023 \cdot \mathrm{Re}^{0.8} \cdot Pr^{0.4} \cdot C_t, \qquad (3)$$

where $\mathrm{Re} = w \cdot d_{eq}/v$ is Reynolds number; $Pr = v/a$ is Prandtl number; C_t is a correction that takes into account the isotherm of the flow.

The correlation is valid for the flow turbulent mode in the range of $10^4 \le \mathrm{Re} \le 10^6$ and $0.7 \le Pr \le 2.0$, the correction that considerers the non-isothermality of the flow for the operating conditions of the first stage can be accepted as $C_t = 1$.

Most of the known literature sources suggest considering the processes of heat and mass transfer of wet steam with a heating surface like for a homogeneous medium. Using the classical correlations in the form of equation (3) leads to inaccuracy of the calculation results, which is related to the inconsistency of the calculation methods of the physical model.

Physical model of two-phase flow formation. We would try to evaluate how justified is the simplification to consider the wet steam as a homogeneous medium when calculating convective heat transfer by refining the physical model of the process.

We will assume that after passing through the channels of the internal space of the SSS from the louvered packages up to the input into the first stage of the superheater, there occurs equalizing of the velocity, temperature, and concentration parameters of the wet steam at the input to the separate cassette of the first stage. The consumption of wet steam through one cassette is

$$D_c = \frac{D_{SSS}}{n_c},$$

where D_{SSS} is steam consumption through one separator-steam superheater of a steam turbine, kg/s; n_c is the number of cassettes of the first stage, pcs.

The total amount of liquid as the drops is determined by the moisture content of the steam when it passes through the louvered packages of the separator

$$D_{lq} = (1 - x_{out}) \cdot D_{SSS}.$$

Unlike a single-phase medium, which can be considered a dry saturated steam, the wet steam contains a certain number of liquid drops depending on $(1 - x_{out})$. The dispersion of the drop size depends on the conditions of their formation, thermophysical properties of liquid and steam. The inertial method of separation, which is used in the SSS, provides a certain dependence of the limiting size of the drops, that will be caught on the geometric characteristics of the louvers, movement parameters and thermophysical properties of the wet steam. It is regulated that the moisture content of a steam should not exceed 1 %. In real conditions, $(1 - x_{out})$may differ several times. The reason for such increasing is the unsatisfactory efficiency of the louvered packages. The primary and secondary removal of liquid drops from the louvered packages facilitate increasing the moisture content $(1 - x_{out})$. Forming the spectrum of drops depends on the factors that determine parameters of forming the primary and secondary removal of drops from the louvered sections.

The primary removal occurs because the design of the louver channel does not ensure the process of catching the liquid drops in the entire range of their size change. It is possible to reduce the amount of primary removal of drops by optimizing the geometry of the louver channel design. Based on the results of researching the movement of liquid drops in curved channels, a correlation was obtained that determines conditions for separating the drops when changing the mode parameters of the two-phase flow and channel geometry [2]

$$\mathrm{Re}_0 = 6.4764 \cdot Bo^{0.25} \cdot We^{0.5} \cdot \left(\frac{\mu''}{\mu'}\right)^{0.15}, \tag{4}$$

where $Bo = \frac{4 \cdot g \cdot (\rho' - \rho'') \cdot r_0^2}{\sigma}$ is Bond number; $We = \frac{2 \cdot \rho' \cdot w^2 \cdot r_0}{\sigma}$ is Weber number; μ' and μ'' are dynamic viscosity coefficients for liquid and vapor, respectively, Pa·s.

The equation (4) is valid in the range of change in Bond number: $6.22785 \cdot 10^{-7} \le Bo \le 9.7989 \cdot 10^{-5}$ and Weber number: $1.025 \cdot 10^{-9} \le We \le 4.48 \cdot 10^{-4}$. Smaller drops will move in curvilinear channels along the complex trajectory, but when the design characteristics of the louver are standard, no contact will occur with the surface, and they will be removed from the louvered package.

The source of secondary removal is the process of plucking the drops from the film surface which is formed on the walls of the louver because of depositing under the action of inertial forces. The film parameters, in particular, its thickness and flow mode depend on the amount of trapped liquid $(x_{b\,in} - x_{out}) \cdot D_{SSS}$, surface tension σ and marginal wetting angle θ, as well as on the balance of forces acting on the liquid film. Unlike the heat and mass exchange devices, where the vapor-liquid flow can move vertically countercurrently or in one direction, in the channels of louver-type separators, the generated liquid film is affected by the forces of gravity, friction and perpendicular tangential stress which is caused by moving the flow of wet steam. As a result, the liquid moves in the film along the complex trajectory: from top to bottom and from the input to the louver channel to its output. A significant part of the film enters the drainage collector and is evacuated from the SSS, but the part of it is collected at the outlet edge of the channel plate as a spindle-shaped stream.

The process of dynamic removal of the drops is characterized by interacting of the flow pulsations, surface tension forces, viscosity, and inertial forces. The conditions for destructing the film or dynamic plucking the drops are determined by processes associated with the Kelvin–Helmholtz instability. At relatively low velocities of the steam flow, the waves on the surface of the liquid film are deformed and destroyed followed by the removal of the drops from the apexes.

Nigmatulin R. B. [1] performed an important analysis of the articles devoted to studying the destruction of films, as well as presenting their results to the conditions of compliance. The given correlations cover a wide range of changes in process parameters. In the articles by Nigmatulin R. B. when determining the limit modes [1] at the input to the measuring section, the temperature, pressure, and consumption of the components of the steam-water or air-water mixture were determined. The tangential stress on the pipe wall τ_w and on the surface of the film τ_{fin} was determined by changing the pressure at the measuring section. At the fixed liquid flow rate at the input to the experimental area, a salt solution was supplied to the wall area, the flow rate of liquid removal from the film D_{cr} was determined by a salt concentration in the liquid samples that were suctioned. It should be noted that by calculating the parameters in the laminar-wave mode, there is a significant deviation of the results of the above studies from the proposed correlation by the amount of ±15 %. Determining the beginning of drop removal by the following correlation is suggested to increase the accuracy of the calculations at $\mathrm{Re} \le 400$

$$We_{13^*} = 3.3205 \cdot Re_\delta^{0.2022} \cdot \mu_{lq} \cdot \left(\frac{g}{\rho_{lq} \cdot \sigma^3}\right)^{0.25}.$$

It is possible to evaluate the results of modelling and physical experiment by the results of Labuntsov D. A. for "capillary wall-liquid" systems with marginal wetting angles $\theta_* = 60$–$140°$. The author proposes to use the Fritz correlation to determine the pre-detaching diameter of the drop [1]

$$d_* = 2 \cdot R_* = 0.0207 \cdot \theta_* \sqrt[3]{\frac{\sigma}{g \cdot (\rho' - \rho'')}}. \tag{5}$$

The size of the drops at the moment of separation can be approximately determined by the Fritz correlation (5) for the process taking place on the edge of the corrugated sheet of the louver-type separator, but to use the thickness of the louvered package plate as a determining size instead of the capillary diameter, or according to the results given in [1].

The authors of [1] use somewhat different approach to determine the limit modes of the beginning of the secondary removal of liquid drops from the wave apexes of the film. The studies accepted the steam flow speed with the smallest value on the correlation curve of the hydraulic resistance coefficient ζ from Reynolds number Re as the critical velocity which determined the beginning of the choking process. Analysing the literature sources on the definition of limit modes proved that using the proposed correlations leads to significant errors. Based on the above, researching the process of disruption of the hydrodynamics of the film flow followed by the subsequent removal of the droplet liquid were performed under conditions corresponding to the operation modes of the louver-type separators. Considering that the wave height depends on the liquid flow rate at the film and the flow mode, research was performed in the laminar-wave mode in the range of $6.25 \le \mathrm{Re}_\delta \le 84.19$. Fig. 1 presents the results of the experimental research.

Generalizing the experimental research results on the disturbing the hydrodynamics of the two-phase flow in the channel made it possible to obtain the correlation of the critical values of the parameters of the two-phase flow to determine the lower limit of the choking process depending on the spray density, the geometric characteristics of the channel, and the physical properties of the liquid and gas with an error of ±7 %

$$K_w = 4 \cdot 10^{-4} \cdot \mathrm{Re}_\delta^{0.5524} \cdot We_{cr}^{1.25},$$

where $K_w = \frac{w''_{cr} \cdot (\rho'')^{0.5}}{(g^2 \cdot \sigma \cdot (\rho' - \rho''))^{0.25}}$ is Kutateladze number; $\mathrm{Re}_\delta = \frac{G}{P_c \cdot \mu}$ is Reynolds film number; $We_{cr} = \frac{\sigma}{(\rho' - \rho'') \cdot D^2}$ is Weber number.

The obtained correlations complete the mathematical description of the hydrodynamics of moving the two-phase medium with the content of drops and a liquid film in the curvilinear channels of the louvered packages of the separation units of the equipment associated with the secondary removal of the liquid.

The main purpose of the article was to bring the methods for calculating the convective heat exchange of the first stage of the SSS into compliance with the physical model of the pro-

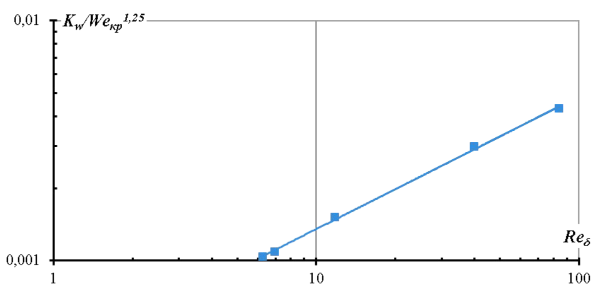

Fig. 1. Stability of gravitational film flow during countercurrent movement of liquid and gas

cess in the case of increasing the steam humidity due to the secondary removal of drops from the separation louvered packages.

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Features of heat exchange of wet steam by forced convection on the pipes with longitudinal finning. A set of numerical studies was performed under limit conditions of the first and the second kind for convective heat exchange for one-phase and two-phase medium under conditions of forced convection.

Modelling the operation of the separation louvered packages made it possible to evaluate the characteristics of the two-phase medium at the input to the first stage of the SSS superheater. The presence of liquid drops in the wet steam significantly changes the kinetics of the heat exchange process between the heat exchange surface as the bundle of vertical tubes with external longitudinal finning and the operating medium. It is possible to consider the following process options:

- the entire range of drops will move with the steam along the heat exchange surface without contacting it;
- the drops will contact the heat exchange pipe and a film of liquid will form on its surface;
- an intermediate case, when a part of the drops will move with the steam, and the other part will form a film.

We will find out for each of the options how much the intensity of heat exchange will depend on the moisture content and the dispersion of the drops by size.

For the first case, it is possible to use the correlation (3) when determining the intensity of heat exchange between the steam and the heat exchange surface if we have a smooth surface. For the finned pipe, using the reduced heat transfer coefficient is recommended, which is determined as follows

$$\alpha_{red} = \alpha_p \cdot E \cdot \frac{F_p}{F_{pw}} + \alpha_{sp} \cdot \frac{F_{sp}}{F_{pw}}, \tag{6}$$

where E is the fin efficiency coefficient; F_p is surface area of all fins, m$^{2;}$ F_{pw} is surface area of the finning wall, m^2; F_{sp} is the surface area of the interfin space, m^2.

As a result of the heat exchange, a symmetrical temperature field is formed in the horizontal cross-section concerning the axis of the channel, which is formed by the finned heat exchange tubes of the cassette located next to them. If it is possible to assume that the liquid drops are uniformly distributed along the horizontal cross-section of the cassette, it is necessary to use the following heat to evaporate them

$$Q_{lq} = D_{lq} \cdot r,$$

where r is the heat of vaporisation at the pressure in the first stage of the SSS, kJ/kg.

Unlike the process of forced convection between the steam and the heat exchange pipe, the liquid drops move together with the steam flow and have a small value of the relative velocity, whose value depends on the balance of mass and inertial forces [3].

A small number of the articles were devoted to studying the process of heat and mass transfer of vapor-liquid flows. They studied the process using numerical modelling or determined the parameters of the process during the physical experiment.

Analysing the physical basis of the heat and mass transfer process during evaporative cooling must be performed considering the gradation of the process of stream breakup and drop deformation [4, 5] and, as a result, the need to consider the behaviour of drops with each other and with the steam flow, as well as the interaction of the generated film with heat exchange surface and vapor phase [6].

The most interesting articles presents the results of studying the processes of heat and mass transfer of two-phase flows in heat exchangers with a finned surface. The main purpose of researching [7, 8] was to improve the characteristics of the equipment in different ways. The ways to increase the coefficients of heat exchange α and mass exchange β [9], to optimise the fin design and to increase the efficiency coefficient E, to reduce the hydraulic resistance ΔP were determined. Furthermore, [8, 10] performed exergy and entropy analysing the processes in the facilities in general. Summarising correlations and efficiency values for heat exchangers with different types of finning were given, but the results from [9, 11] are not sufficiently correlated with each other, it is explained due to the different design of the heat exchanger.

The impact of the drop liquid on the intensity of processes was studied. The articles [12, 13] considered the case when the film is distributed evenly along the heat exchange surface, and the friction force at the "gas–liquid" boundary is a small value, which does not correspond to real conditions. Analysing such case must consider the value of the minimum spray density $\mathrm{Re}_{\delta\,\mathrm{min}}$, the marginal wetting angle θ, and the value of the surface tension σ. Considering these parameters will make it possible to estimate the real area of the wetting surface of the heat exchanger.

By using the staged approach to consider the process in the superheated part of the SSS, it is possible to improve the accuracy of calculations. It is recommended to use correlation [1] to determine the intensity of the drop evaporation process:

- for heat exchange

$$Nu = 2 + 0.6 \cdot \mathrm{Re}^{0.5} \cdot Pr^{0.33};$$

- for mass exchange

$$Nu_D = 2 + 0.57 \cdot \mathrm{Re}^{0.5} \cdot Pr_D^{0.33},$$

the range of Reynolds number change is $0 \le \mathrm{Re} \le 200$, and the range of Prandtl number corresponds to $0.6 \le Pr \le 1{,}000$.

Thus, the intensity of heat and mass exchange processes between the liquid drop and a steam is determined using the methods of numerical modelling or correlation obtained from the results of a physical experiment. Considering the insignificant value of the temperature drop due to the initial and boundary conditions, the specific mass flow from the drop surface is determined as

$$j_{ic} = \frac{\alpha \cdot \Delta t}{r},$$

where Δt is temperature drop between drop surface and steam, which is heated, °C.

Assuming that the drops have the shape of a sphere, the time of complete evaporation of the drops can be determined as

$$\tau = 10.667 \cdot \frac{d_0 \cdot \rho'}{j_{ic}},$$

where d_0 is drop diameter at the input to the first stage of the superheater, m; ρ' is liquid density, kg/m^3.

The obtained time of evaporation of liquid drops must be compared with the residence time of the dispersed phase in the superheater stage

$$\tau^* = \frac{L \cdot \rho'' \cdot F_c}{D_c},$$

where L is the length of the cassette, m; F_c is live cross-sectional area of the cassette, m^2; ρ'' is vapor density at average temperature, kg/m^3; D_c is mass flow rate of steam through the cassette, kg/s.

Comparing the values τ and τ^* makes it possible to evaluate the steam state at the output from the first stage of the superheater.

In other case when liquid drops contact the surface of the heat exchange finned tubes of all cassettes of the first stage of the superheater, it is necessary to consider the fact that the liquid film is formed on the surface, and this liquid film creates additional thermal resistance to the process of heat transfer from the tube to the heated steam. On the other hand, the coefficient of heat transfer from the pipe wall to the steam according to correlation (3) will have a smaller value than the value that the coefficient of heat transfer from the wall to the liquid film, which is determined according to the correlation obtained by B. G. Ganchev [1]

$$Nu = 0.01 \cdot \mathrm{Re}_\delta^{0.83} \cdot Pr^{0.33},$$

where Re_δ is Reynolds film number.

Accepting that in the presence of the dynamic influence of the steam flow, the thickness of the film formed from liquid droplets will be within the limits close to the minimum values of the film Reynolds number. The correlation, which is valid for smooth and submerged surfaces with a capillary-porous coating met such conditions [1, 14]

$$\mathrm{Re}_{\delta\,\min} = \mathrm{Re}_{\delta\,\min 0} + 0.31 \cdot \mathrm{Re}^{0.4},$$

where $\mathrm{Re}_{\delta\,\min 0}$ is minimal Reynolds number of film, is determined by the amount of liquid that enters the cassette of the first stage of the superheater with the steam

$$\mathrm{Re}_{\delta\,\min 0} = \frac{D_c \cdot (1 - x_{out})}{P_c \cdot \mu_{lq}},$$

where P_c is the cross-sectional perimeter of all heat exchange pipes from the outside, m.

The minimum thickness of the film under such conditions of its formation is determined as

$$\bar{\delta}_{\min} = \left(\frac{3 \cdot \mathrm{Re}_{\delta\,\min} \cdot v_{lq}^2}{4 \cdot g} \right)^{1/3}.$$

The area of the heat exchange finned surface on which the film of liquid was formed is defined according to the moisture content of the wet steam $(1 - x_{out})$, the amount of liquid that enters the cassette and the minimum thickness of the film. Further thermal calculation is performed based on the average value of the heat transfer coefficient, considering the dry and wetting surface.

The considered limit modes of heat exchange of the two-phase flow with the finned surface of the tubes of the first-stage superheater cassette cannot be fully implemented at the real equipment, but they provide an opportunity to compare the calculated values with the real parameters of the two-phase environment to determine ways to improve the operating the SSS.

An important aspect, which must be considered when choosing the ways to modernize the external heat exchange surface, is optimizing the finning design. Structurally, the heat exchange pipe has the following dimensions: the diameter $d_{out} \times \delta_w = 16 \times 2$ mm and the length $L = 4{,}620 - 4{,}750$ mm. The fins made of a П-shaped profile with a thickness $\delta_p = 0.8$ mm are welded on the outer surface along the pipe. The total number of fins is $n_p = 12$ pcs. The heat exchange pipe is made of grade 20 carbon steel.

It is possible to evaluate the perfection of the design of the heat exchange surface by modelling the heat exchange process when changing the parameters of the heat carrier and the operating medium, as well as the geometric characteristics of finning of the heat exchange pipe, namely, the height of the fin h_p, the thickness of the fin δ_p. According to dependence (6), the impact of the geometric dimensions of the fin on the reduced heat transfer coefficient α_{red} considers the coefficient of the fin efficiency E. The coefficient of the fin efficiency is determined as

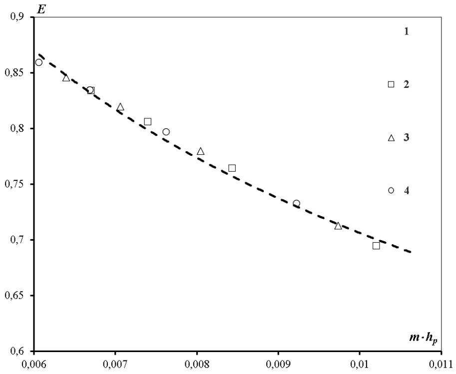
$$E = \frac{th(m \cdot h_p)}{m \cdot h_p},$$

where $m \approx \sqrt{\dfrac{2 \cdot \alpha}{\delta_p \cdot \lambda_p}}$ is fin parameter, 1/m.

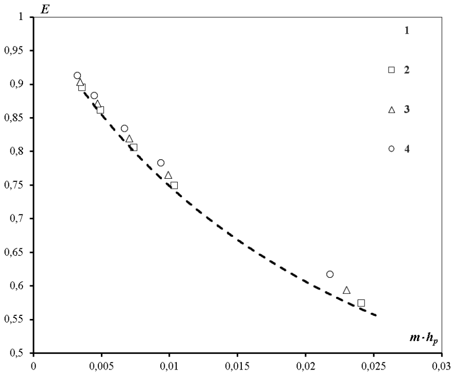
Modelling the heat exchange process was performed by changing the geometric characteristics of the fin in a wide range: the fin height $h_p = 6–15$ mm of the fin thickness $\delta_p = 0.4–1.0$ mm. Considering the possible change in the load of the power equipment, the effectiveness of finning was checked in the range of thermal power $Q = 100–70$ %. Fig. 2 presents the results of numerical modelling. It should be noted that coefficient of the fin efficiency E is a decreasing function by growing the complex $(m \cdot h_p)$ in the entire range of the loads. Based on the analysis of the results, the following correlations were obtained:

- to determine the impact of the fin height

$$E = 0.0834 \cdot (m \cdot h_p)^{-0.462}; \quad (7)$$



a



b

Fig. 2. Dependence of the fin efficiency coefficient on the product of the fin parameter and its height at different thermal loads:

a – from the fin thickness; b – from the fin height; 1 – correlation; 2 – 90 % load from nominal; 3 – 80 %; 4 – 70 %

- to determine the impact the fin thickness

$$E = 0.0834 \cdot (m \cdot h_p)^{-0.462}. \quad (8)$$

Thus, the correlations are obtained for pipes with longitudinal finning in the form of the Π-shaped profile, welded to the outer surface of the pipe. Based on such correlations it is possible to optimize the geometric characteristics of the heat exchange surface. Solving this problem requires a complex approach taking into consideration the nonlinear nature of the obtained correlations (7, 8) and the ambiguous impact of changing the size of the fin on α_{red}. Changing the height and thickness of the fin leads to changing the efficiency coefficient by 30.6 and 24.9 %, respectively, with all other fixed characteristics.

No less important factor, except for the analysed ones, which directly affects the value of the heat transfer coefficient k_{l1} (2), is the thermal resistance that occurs when disturbing the contact of the welding joint of the Π-shaped fin profile with the outer surface of the pipe R_2. The technology of manufacturing the heat exchange pipe involves welding the Π-shaped profile to the outer surface of the pipe by means of contact welding with a continuous or discontinuous seam. When conducting diagnostics of power equipment, it is necessary to consider not only the features of the design, the manufacturing technology, but also the operation period and conditions. In conditions of sign-changing stresses associated with changing the load, the features of the operation on the wet steam and the equipment vibration, disturbing the integrity of the welding joint between the fin and the heat exchange pipe is a characteristic consequence of these factors. Increasing the thermal resistance of the contact R_2 deteriorates the heat transfer process by 10–20 % [15], the real value can be determined by the results of testing and diagnosing the equipment.

Conclusions and recommendations. In the article, the authors performed a comprehensive analysis of the design features and operating conditions of the cassette of the first stage of the separator-steam superheater SSS, located between the HPC and LPC of the VVER-1000 steam turbine, which operates on the wet steam.

The reasons for the non-compliance of the steam parameters after the first stage with the design values are related to the removal of drops from the surface and the rear edge of the plates of the louver-type separator. The above-mentioned factors lead to occurring of the secondary removal and the growing the steam humidity $(1 - x_{out})$ above the design values.

Generalising the results of experimental research of disturbing the hydrodynamics of the two-phase flow movement made it possible to obtain the correlation of the critical values of the two-phase flow parameters for determining the lower limit of the process of the drop plucking from the liquid film depending on the irrigation density, the geometric characteristics of the channel, and the physical properties of the liquid and gas with an error of ±7 %.

The heat exchange process is complicated by the presence of the liquid drops in the two-phase flow. To determine the coefficient of heat transfer from the outer surface of the finned pipe to the wet steam, it is necessary to consider its humidity at the input to the cassette and the size of the drops. Standard methods for determining the intensity of heat exchange during the movement of the operating medium along the surface suggest using the correlation for a homogeneous medium, that significantly reduces the accuracy of the obtained results.

The considered limit modes of heat exchange of the two-phase flow with the finned surface of the tubes of the first-stage superheater cassette cannot be fully implemented at the real equipment, but they provide an opportunity to compare the calculated values with the real parameters of the two-phase environment to determine the ways to improve the operation of the SSS.

For pipes with longitudinal finning in the form of an Π-shaped profile welded to the outer surface of the pipe, the correlations were obtained, based on which it is possible to optimise the geometric characteristics of the finning. Solving this problem requires a complex approach, considering the nonlinear nature of the obtained correlations and the ambiguous effect of changing the dimensions of the fin on the reduced heat transfer coefficient.

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Теплообмін при поздовжньому русі вологої пари в оребрених теплообмінниках

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Робота присвячена дослідженню гідродинаміки й теплообміну двофазних середовищ. При проєктуванні технологічного обладнання, в якому в якості робочого тіла використовується волога пара, у більшості випадків не враховуються особливості взаємодії між краплями рідини й теплообмінною поверхнею. У повній мірі це стосується парових турбін, що працюють на вологій парі, вологовміст якої залежить від первинного та вторинного виносу крапель рідини з сепараційних блоків.

Мета. Удосконалення методики розрахунку рекуперативних теплообмінних апаратів, де в якості робочого тіла використовується волога пара.

Методика. Заснована на аналізі фізичної моделі руху двофазного середовища в умовах тепломасообміну з урахуванням конструктивних характеристик поверхні теплообміну.

Результати. Отримана кореляція критичних значень параметрів двофазного потоку для визначення нижньої границі процесу зриву крапель із плівки рідини в залежності від густини зрошування, геометричних характеристик каналу й фізичних властивостей рідини та газу. Для труб із повздовжнім оребренням у вигляді П-подібного профілю отримані кореляції, на підставі яких рекомендується виконувати оптимізацію геометричних характеристик оребрення.

Наукова новизна. Визначення граничних режимів утворення вторинного виносу при русі двофазного середовища в сепараційних пристроях і особливостей тепломасообміну вологої пари в оребрених рекуперативних теплообмінниках.

Практична значимість. Представлені результати дозволяють виконати оптимізацію конструкції рекуперативних теплообмінних апаратів із повздовжнім *п*-подібним оребренням.

Ключові слова: *гідродинаміка руху, двофазний потік, рекуперативний теплообмінний апарат, волога пара*

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