УДК 621.184.64:62-9+65.011.46 DOI: https://doi.org/10.15407/geotm2021.156.062

CALCULATION OF THERMOHYDRAULIC EFFECTIVENESS OF THE POROUS ONCE-THROUGH WATER STEAM GENERATORS IN LAMINAR FLOW REGION WITH BOUNDARY CONDITIONS OF THE FIRST KIND

[1]Lukisha A.P.
[1]Institute of Geotechnical Mechanics named by N. Poljakov of NAS of Ukraine

РОЗРАХУНОК ТЕПЛОГІДРАВЛІЧНОЇ ЕФЕКТИВНОСТІ ПОРИСТИХ ПРЯМОТОЧНИХ ВОДЯНИХ ПАРОГЕНЕРАТОРІВ В ЛАМІНАРНІЙ ОБЛАСТІ РУХУ ТЕПЛОНОСІЯ І ПРИ ГРАНИЧНИХ УМОВАХ ПЕРШОГО РОДУ

[1]Лукіша А.П.
[1]Інститут геотехнічної механіки ім. М.С. Полякова НАН України

РАСЧЁТ ТЕПЛОГИДРАВЛИЧЕСКОЙ ЭФФЕКТИВНОСТИ ПОРИСТЫХ ПРЯМОТОЧНЫХ ВОДЯНЫХ ПАРОГЕНЕРАТОРОВ В ЛАМИНАРНОЙ ОБЛАСТИ ДВИЖЕНИЯ ТЕПЛОНОСИТЕЛЯ И ПРИ ГРАНИЧНЫХ УСЛОВИЯХ ПЕРВОГО РОДА

[1]Лукиша А.П.
[1]Институт геотехнической механики им. Н.С. Полякова НАН Украины

Abstract. The paper presents results of calculations of thermohydraulic efficiency of the porous once-through steam generators with water as a model working fluid in the region of the heat-transfer agent laminar motion and with boundary conditions of the first kind. The smooth-wall cylindrical channels with different diameters were used as the reference surfaces to be compared. The following operating and design parameters were taken as a calculation base: temperature of the liquid on the saturation line at entry into the channel was: T_{so} = 280 °C, 300 °C, 320 °C, 340 °C, 360 °C. Temperature head, i.e. a difference between the wall temperature and temperature of the liquid at entry into the channel was: $\Delta T=T_w-T_o$ = 1 °C; 2 °C; 3 °C; 4 °C; 5 °C. The Reynolds number at entry into the channel was: Re_o = 700; 1000; 1200; 1500; 1700; 2000; 2300. The channel porosity was: θ = 0.7; 0.75; 0.8; 0.85; 0.9. The porous material was metal felt with the copper fiber diameter of 200 microns. The channel diameter was: d =$3\cdot10^{-3}$ m; $4\cdot10^{-3}$ m; $5\cdot10^{-3}$ m; $6\cdot10^{-3}$ m; $7\cdot10^{-3}$m; $10\cdot10^{-3}$ m. On the basis of the performed computational studies, it was concluded that for the conditions of the same mass flow rates of the coolant, with laminar mode of motion, and the same channel diameters, it is possible to achieve a significant reduction in the length of the porous once-through steam generator in comparison with the length of the smooth-wall once-through steam generator; however, under these conditions it is not possible to reduce pressure drop in the channels and, accordingly, to reduce power consumption needed for heat-transfer agent pumping. This computational study also made it possible to establish main regularities in behavior of the energy efficiency coefficients and their dependence on the model operating and design parameters.

Keywords: thermo-hydraulic efficiency; porous steam-generating channels; heat-transfer agent, water; boundary conditions of the first kind; laminar flow regime.

Intorduction. The limitedness of energy resources in the world require to find new energy-saving technologies. One of the ways in this field is to study a possibility to use energy-efficient porous high-thermal-conductivity channels during phase transition of the heat-transfer agent evaporation, i.e. porous steam generators. Since water is the most common heat-transfer agent in industrial heat engineering installations and devices, we are investigating the issue of thermohydraulic efficiency of the porous steam generators for this type of heat-transfer agent. It is known from the literature that most of the thermophysical studies of the issues of the boiling and evaporation phase transition (in smooth-wall and porous channels) were carried out for boundary conditions of the second kind, i.e., at a constant heat flow on the channel wall. However, as the study of the issue of thermohydraulic efficiency of porous channels for a single-phase heat-transfer agent has shown, the energy effect can be obtained at boundary conditions of the first or third kind, i.e. either at constant temperature on the channel wall or if the law of heat transfer with a known ambient temperature acts.

Let's consider the problem and try to find thermal-hydraulic efficiency of the porous cylindrical channels at the phase transition of heat-transfer agent-water evaporation on the boundary conditions of the first kind. As a reference surface for the comparison, we choose a smooth-wall tube operating as a once-through steam generator. Below are the calculated ratios used in the study and calculations.

Boiling in the smooth-wall cylindrical channels (pipes). Zone of nucleate boiling. While creating the methodology for calculating thermal characteristics in the smooth and porous channels, we assume that a liquid at entry into the channel is on a saturation line. According to [1], heat transfer coefficient in zone of the developed nucleate boiling is defined by the formula

$$\alpha = \alpha_1 \sqrt{1 + 7 \cdot 10^{-9} (\rho w_{mix}\, r/q)^{3/2} (0{,}7\alpha_o/\alpha_1)^2}, \qquad (1)$$

where

$$\alpha_1 = \sqrt{\alpha_к{}^2 + (0{,}7\alpha_o)^2}, \qquad (2)$$

here $\alpha_к$ is heat transfer coefficient for the one-phase water flow in the tube or channel, which calculated by the formula

$$\mathrm{Nu}_d = \frac{\alpha_к d}{\lambda} = \frac{(\xi/8)\mathrm{Re}\cdot\mathrm{Pr}\cdot C_t}{k + 4{,}5\sqrt{\xi}\quad(\mathrm{Pr}^{2/3} - 1)}, \qquad (3)$$

where

$$k = 1 + 900/\mathrm{Re}; \qquad (4)$$

$$\xi = (1{,}82\, lg\, \mathrm{Re} - 1{,}64) \qquad (5)$$

C_t is correction for the flow nonisothermality.

For liquid drops at $\mu_w/\mu_t = 0{,}08 \div 40$

$$C_t = (\mu_f/\mu_w)^n, \qquad (6)$$

where $n = 0{,}11$ when liquid is heated; $n = 0{,}25$ when liquid is cooled.

In formula (6), μ is a coefficient of dynamic viscosity, indices f and w relate to the mean temperature and the wall temperature, respectively.

Value α_o in formula (1) is the heat transfer coefficient at boiling in a large volume. The value α_o is calculated by formula

$$\alpha_o = 4{,}34 q^{0,7} (p^{0,14} + 1{,}35 \cdot 10^{-2}\, p^2), \qquad (7)$$

where q is value of the specific heat flow, W/m^2; p is pressure, MPa; α_o is the heat transfer coefficient, W/(m^{2}K).

Value w_{mix} in formula (1) is an average speed of the water-steam mixture flow, m/s. It is calculated by formula

$$w_{mix}=w_o[1+x(\rho'/\rho''-1)], \tag{8}$$

where w_o is flow speed, m/s; ρ' and ρ'' are density of the liquid and vapor on the saturation line, respectively; x is expendable mass vapor content.

As it is seen, in order to determine the heat transfer coefficient, the value α of the heat flow q is introduced into the formula (1). When value $q=\alpha(t_w - t_s)$ is used in the equation (1), one can see that in this case value α is included into the left and right sides of the equation. It means that equation (1) is non-linear algebraic equations for the heat flow α, which cannot be solved analytically in an explicit form. For calculating the value α by equation (1), a method of consecutive approximations should be used.

Calculation of boiling heat transfer in cylindrical smooth wall channels in the dryout zone. When liquid is boiled in the smooth-wall tube with increasing value of the mass consumable steam-content until reaching a certain level called a boundary vapor content, a dispersed flow regime occurs at which a film of the liquid on the wall starts drying. The flow being a mixture of steam and vaporized liquid droplets begins to move. During this process, heat transfer inside the channel significantly drops. According to S.S. Kutateladze [2], for the purpose of calculating the boundary steam- content, this dependence can be represented in the following form

$$x_{\text{bound.}}=0{,}3+0{,}7\exp(-45\tilde{w}) \tag{9}$$

where $\tilde{w}=[\rho w\mu'/(\sigma\rho')](\rho/\rho'')$; $\rho w = w_g$ is mass velocity .

The heat transfer coefficient in the dryout zone was calculated from the equation proposed by Z. L. Miropol'skii [3] :

$$\mathrm{Nu}'' = 0{,}028\mathrm{Re}''^{0,8}\mathrm{Pr}''^{0,4}(\rho'/\rho'')^{1,15} , \tag{10}$$

where $\mathrm{Nu}'' = \alpha d/\lambda''$; $\mathrm{Re}'' = w''d/\nu''$ is Reynolds number calculated by the reduced speed of steam $w_r = (\overline{\rho w})x/\rho''$. Here d is diameter of the channel; λ'' is thermal conductivity of steam.

Calculation of heat transfer during motion of the two-phase flow in porous materials. The problem of experimental study of heat transfer when a steam-water mixture moves through the porous media was considered by Kalmykov I. V. in his dissertation [4]. Basing on the summarized experimental data, the author obtained the following generalizing dependence, which characterizes heat exchange when steam-water mixture moves through the porous media

$$\alpha_V / \alpha_{VLO} = 1 + 810x\sqrt{(\rho w)_0 \frac{\nu'}{\sigma} \cdot \frac{1-x}{x}}\ , \tag{11}$$

where α_V is a coefficient of volume pore heat exchange when two-phase steam-liquid flow passes through the porous material, W/m^{3}K; α_{VLO} is a heat exchange coefficient of volume pore heat exchange (an amount of heat absorbed by the fluid per unit time per unit volume of the porous body) when liquid moves through the porous material at a rate equal to the total mixture flow, W/m^3·degree. $(\rho w)_0=\dot{m}/F_{\Pi}$ is a rate of mixture filtration (mass flow rate, kg/m^{2}s; $\dot{m}$ is total mass flow rate of mixture, kg/s; $F_{cs}=\pi d^2/4$ is a cross-sectional area of the channel, m^2; d is a diameter of the channel, m; ν' is a coefficient of kinematic viscosity of the liquid on the saturation line, m^2/s; σ is a coefficient of the liquid surface tension, N/m; x is a mass expendable of steam content

In order to find value of α_{VLO}, the recommended by Kostornov A.G. [5] dependence was used for calculating heat transfer inside the porous metal-fibre material when it moves through the one-phase heat-transfer agent

$$\mathrm{Nu}=0{,}007\mathrm{Re}^{1{,}2} \tag{12}$$

where $\mathrm{Nu_V} = \alpha_V(\beta/\alpha)^2/\lambda_l$, $Re = \dfrac{(\rho w)_0(\beta/\alpha)}{\mu_l}$, index l refers to the parameters of liquid.

As it is seen from the expression (11), it does not contain parameters of the porous structure characteristic in their explicit form, such as particle size, or flow resistance coefficients α and β. This makes it possible to use these expressions for almost any porous materials.

According to [4], one of the minor drawbacks of the formula (11) is that in the course of the experiments with high steam contents thermal nonequilibrium of the flow occurred. Therefore, generalization of the experimental data was performed only for those values α_v, which corresponded to $x \leq 0{,}8$. It was not recommended to apply formula (11) when steam content was more than 0.8 as this could lead to a significant error. Though it was recommended to estimate value α_v in the interval of $x \approx 0{,}8 \div 1{,}0$ by interpolation of values α_v calculated by (11) at x = 0.8 and by expressions characterizing the convective heat transfer in porous material with steam heat-transfer agent flow (x = 1) [5].

$$Nu'' = 0{,}007 Re''^{\,1{,}2}\ , \tag{13}$$

where
$$Nu'' = \frac{h(\beta/\alpha)^2}{\lambda''}\ ;\ \ Re'' = \frac{G\ (\beta/\alpha)}{\mu''}\ , \tag{14}$$

where Nu″ and Re″ are the Nusselt number and Reynolds number, respectively, calculated by thermal properties of steam on the saturation line; $G = (\rho w)_0$ is full mass flow rate of the heat-transfer agent, kg/m^{2}s.

In calculations of specific heat flow, which is absorbed by the heat-transfer agent during evaporation in a porous cylindrical channel, expression obtained in [6] was used

$$q = (\lambda h_V)^{1/2} \frac{I_1(\gamma)}{I_0(\gamma)} \vartheta(1) \ , \tag{15}$$

where $I_0(\gamma)$ and $I_1(\gamma)$ are the modified Bessel functions of the first kind of zero and first order, respectively; λ is a coefficient of porous material thermal conductivity, which should be calculated with taking into account material of a fuse frame and liquid and steam thermal conductivity on the saturation line; h_V is intensity of the volume pore heat transfer; $\gamma^2 = \frac{h_V (d/2)^2}{\lambda}$ is parameter, which characterizes intensity of the pore heat exchange; d is a diameter of channel; $\vartheta = T - t_s$, where T is temperature of the porous frame; t_s is the coolant temperature on the saturation line; $\vartheta(1) = T_w - t_s$, is temperature difference of the porous frame and the cooler on inner surface of the wall.

Calculation of Hydraulic Resistance of the Smooth-Wall Channels to Motion of Two-Phase Steam-Liquid Flow. For calculating hydraulic resistance of the tubes and channels to motion of two-phase steam-liquid flow, the method developed by Lockhart and Martinelli [7] is commonly used. The essence of the method is that due to friction of the two-phase flow, the pressure gradient is usually expressed by coefficients, which are multiplied by the respective gradients of the single-phase flows, i.e.

$$\left(\frac{dp}{dz}\right)_{tw.ph.} = \Phi_L^2 \left(\frac{dp}{dz}\right)_L \tag{16}$$

or

$$\left(\frac{dp}{dz}\right)_{tw.ph.} = \Phi_G^2 \left(\frac{dp}{dz}\right)_G \ , \tag{17}$$

where $\left(\frac{dp}{dz}\right)_{tw.ph.}$ is pressure drop caused by friction of the two-phase flow; $\left(\frac{dp}{dz}\right)_L$ and $\left(\frac{dp}{dz}\right)_G$, are the pressure drop for a liquid or gas, respectively, if the liquid or gas occupied the entire cross section of the tube; Φ_L^2 and Φ_G^2 are empirically determined coefficients; z is the coordinate.

Lockhart and Martinelli [7] found that the coefficients Φ_L^2 and Φ_G^2 are the function of the parameter X^2, which is determined by the following way

$$X^2 = (\frac{dp}{dz})_L \Big/ (\frac{dp}{dz})_G \ . \tag{18}$$

Chisholm and Sutherland [8] proposed an analytical expression for the dependences $\Phi_{L,G}^2 = f(x)$ in the following form

$$\Phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \ , \tag{19}$$

$$\Phi_G^2 = 1 + CX + X^2 \ , \tag{20}$$

where
$$C = \frac{1}{K}\sqrt{\frac{\rho_L}{\rho_G}} + K\sqrt{\frac{\rho_G}{\rho_L}} \ . \tag{21}$$

where K is the slide factor.

Zivi, basing on a simple model [9], found that for the steam-water flow

$$K = (\rho_L/\rho_G)^{1/3} \ . \tag{22}$$

Having analysed the correlations (16) - (22) it becomes apparent that this system of equations allows calculating pressure drop in the channels and tube at two-phase motion of steam-liquid flow.

In order to calculate the parameter $X^2 = (\frac{dp}{dz})_L \Big/ (\frac{dp}{dz})_G$, the correlations are used which characterize pressure drop in a single-phase flow at different velocities (Reynolds numbers) of the cooler motion [10].

In the general case, the pressure drop across the pipe at the motion of a single-phase liquid or gas (vapour) heat carrying agent is calculated as per the formula [10]

More generally, pressure drop in the tube at the motion of a single-phase liquid or gas (steam) heat-transfer agent is calculated by formula [10]

$$\Delta P = \xi \cdot \frac{l}{d} \cdot \frac{\rho u_{av}^2}{2} \ , \tag{23}$$

where ξ is a coefficient of tube resistance; l, d are length and diameter of the tube, correspondingly, m; ρ is a density of the liquid (gas), kg/m^3; u_{av} is an average velocity of liquid along the cross section in the tube, m/s.

With taking into consideration that the Reynolds number is calculated from the correlation $Re = \frac{\rho u_{av} d}{\mu'}$, the expression (23) can be transformed into

$$\Delta P = \xi \cdot \frac{l}{d} \cdot \frac{\rho \left(\frac{Re\mu'}{\rho d} \right)^2}{2} . \qquad (24)$$

For laminar flow when the Reynolds number is Re ≤ 2300, the coefficient of resistance is determined from the correlation [10]

$$\xi = 64/\mathrm{Re_o} \qquad (25)$$

In order to calculate the coefficients of hydraulic resistance for areas with transitional and turbulent flow in a smooth-wall tube, we used the correlations developed by S. S. Kutateladze [11],

$$\xi \approx 6{,}3\ 10^{-3}\ \mathrm{Re}^{0{,}5} \qquad (26)$$

by Blasius [10]

$$\xi = 0{,}3164\ \mathrm{Re}^{-1/4} \qquad (27)$$

and Nikuradze [10]

$$\xi = 0{,}0032 + 0{,}221/\mathrm{Re}^{-0{,}237} \qquad (28)$$

When values $(\frac{dp}{dz})_L$ and $(\frac{dp}{dz})_G$ are calculated by formulas (16) and (17), the following values of Reynolds numbers should be used:

$$\mathrm{Re_L} = \mathrm{Re_o}\,(1 - x)\ , \qquad (29)$$

$$\mathrm{Re_G} = \mathrm{Re_o}\, x\, (\mu'/\mu'')\ , \qquad (30)$$

where $\mathrm{Re_o}$ is the Reynolds number at entry into the channel; x is the mass expendable steam-content of the flow; μ' and μ'' are coefficients of dynamic viscosity of liquid and steam, correspondingly, on the saturation line.

Calculation of hydraulic resistance of the porous high-thermal-conductivity channels to the motion of a two-phase steam-liquid heat-transfer agent. The hydraulic resistance of the porous high-thermal-conductivity channels to the motion of a two-phase steam-liquid heat-transfer agent is calculated by the method similar to the method developed by Lockhart-Martinelli's [7]. According to this method, which Yu. A. Zeigarnik and I.V. Kalmykov applied to the motion of two-phase steam-liquid adiabatic flows in porous media [4], [12], resistance of porous channels to the friction of adiabatic two-phase mixture flow is calculated by the formulas:

$$(\frac{\Delta P}{\ell})_{tw.ph.\ por.} = (\Delta P / \ell)_L \cdot \Phi_L^2\ , \qquad (31)$$

or

$$\left(\frac{\Delta P}{\ell}\right)_{tw.\ ph.\ pr.} = (\Delta P/\ell)_G \cdot \Phi_G^2\ , \tag{32}$$

where $(\Delta P/\ell)_\mathrm{L}$ and $(\Delta P/\ell)_\mathrm{G}$ are pressure drops if through the porous structure only the liquid phase flows in the amount contained in the mixture, or only the steam phase. These values are calculated by the equation (33), in which the corresponding mass velocities of each phase filtration are used as mass filtration velocities $\rho w = G$.

$$\frac{-dP}{dZ} = \alpha\mu\upsilon G + \beta\upsilon G^2 \tag{33}$$

where α and β are the inertia and viscosity coefficients of the porous material resistance; μ and υ are dynamic viscosity coefficient and specific volume of liquid; G is the liquid mass flow rate through the porous material.

According to [7], parameters Φ_L and Φ_G are the function of Martinelli parameter $X = \sqrt{(\Delta P/\ell)_\mathrm{L}/(\Delta P/\ell)_\mathrm{G}}$. The following formula can be written for X^2 parameter [12] :

$$X^2 = \left(\frac{1-x}{x}\right)\left(\frac{\mu'}{\mu''}\right)\left(\frac{\rho''}{\rho'}\right)\frac{1+(\beta/\alpha)[G(1-x)]/\mu'}{1+(\beta/\alpha)[G\cdot x]/\mu''} = \left(\frac{1-x}{x}\right)\left(\frac{\mu'}{\mu''}\right)\left(\frac{\rho''}{\rho'}\right)\frac{1+\mathrm{Re}'}{1+\mathrm{Re}''} \tag{34}$$

The interrelation of the parameter X with Φ_L and Φ_G can be expressed analytically [8]

$$\Phi_L^2 = 1 + C/X + 1/X^2\ , \tag{35}$$

$$\Phi_G^2 = 1 + CX + X^2\ . \tag{36}$$

Having processed the test data, Yu. A. Zeigarnik and I.V. Kalmykov [12] concluded that coefficient C is the function of mass filtration velocity $(\rho w)_0 = G$, physical characteristics (ρ' and μ') and hydraulic characteristics of porous structure (coefficients α and β)

$$C = 4{,}0\left(\frac{\beta/\alpha}{G}\cdot\frac{\rho'}{\mu'}\right)^{0,4}. \tag{37}$$

Procedure for calculating thermophysical parameters of a cylindrical channel when a two-phase steam-liquid flow moves through it with the boundary conditions of the first kind. While considering the issue of thermohydraulic efficiency of porous channels, the following feature should be kept in mind. The comparison of the channel characteristics should be performed with the boundary conditions of the first or third kind, i.e., at a given constant wall temperature or at a given law of heat transfer on the outer surface of the channel and given ambient temperature. Only with these two types of boundary conditions it is possible to expect certain positive thermohydraulic effect of porous channels in comparison with the smooth-wall channels. When specifying boundary conditions of the second kind, i.e. when setting a rate of the constant specific heat flow on the outer surface of the channel, there should be no thermophysical effect, since in this case, according to the law of conservation of energy, both the smooth-wall and the porous channels having the same length absorb the same amount of heat and, therefore, evaporate the two-phase flows with same steam content. In this article, the calculation of the thermohydraulic efficiency with the boundary conditions of the first kind is considered. Since it is known from the literature that most of the thermophysical studies on the boiling and evaporation phase transitions (in smooth-wall and porous channels) were carried out with boundary conditions of the second kind, i.e., with a constant amount of specific heat flow on the channel wall, then, in order to calculate thermal-hydraulic efficiency of the porous channels with boundary conditions of the first kind, it is necessary to recalculate the initial equations with boundary conditions of the second kind for the boundary conditions of the first kind. A full description of this problem includes a fairly great amount of material and is not in a research focus of this article. More details concerning this method applied for the smooth-wall channels can be found in [13], and for the porous channels in [14].

Results of the work. As a result of the work performed, calculations of the porous channel efficiency for evaporation of the steam-water flow were conducted. A smooth-wall tube was used as a reference surface for the comparison. The following operating and design parameters were taken as a calculation base: the temperature of the liquid on the saturation line at entry into the channel was T_{so}=280 ^{o}C, 300 ^{o}C, 320 ^{o}C, 340 ^{o}C, 360 ^{o}C. Temperature head, i.e. a difference between the wall temperature and temperature of the liquid at entry into the channel was $\Delta T=T_w-T_o$ = = 1, 2, 3, 4, 5 ^{o}C. The Reynolds number at entry into the channel was: Re_o = 700, 1000, 1200, 1500, 1700, 2000, 2300. The channel porosity was: θ = 0,7; 0,75; 0,8; 0,85; 0,9. The porous material was metal felt with the copper fiber diameter of 200 microns. The channel diameter was: d =$3\cdot10^{-3}$; $4\cdot10^{-3}$; $5\cdot10^{-3}$; $6\cdot10^{-3}$; $7\cdot10^{-3}$; $10\cdot10^{-3}$ m. Totally, 4,200 design points characterizing the model operating and design parameters were calculated.

The plots of the calculated dependences for the geometric factor of the porous channel efficiency k_F, ratio of pressure drops $k_P=\Delta P_{por}/\Delta P_{sm}$, parameters Q/N for the porous and smooth-wall channels, as well as their lengths as a function of the Reynolds number and porosity f(Re, Θ) for inlet temperature of liquids T_s=360 ^{0}C are shown in Figures 1–6, respectively. Here, Q is an amount of heat received by the channel; N is a power consumption needed for pumping the heat-transfer agent. Behavior of these parameters within the range of the inlet liquid temperature T_s=280-360 ^{0}C (with porosity of the porous channel θ=0,9) is shown in Figures 7-12. Figures 13 and 14 show the behavior of coefficient of the porous channel efficiency k_F = l_{sm}/l_{por} and ratio $k_P=\Delta P_{por}/\Delta P_{sm}$ depending on the change in the Reynolds number of the flow Re_0 at entry into the channel and the change in the temperature difference $\Delta T=T_{wall}-T_{s0}$, where T_{s0} is a liquid temperature at entry into the channel. These plots were built with the following parameter values: porosity Θ = 0,9; channel diameter d = 0.004 m. The liquid temperature on the saturation line at entry into the channel was T_{s0} =360^{0}C. Fig. 15 and 16 show the similar plots, but with liquid temperature

T_{s0}=300^{0}C at entry into the channel. Figures 17 and 18 show, respectively, behavior of the efficiency coefficient k_F and the ratio $k_P=\Delta P_{por}/\Delta P_{sm}$ depending on the channel diameter *d* and the Reynolds number Re. In the issue under consideration, rate of the heat-transfer agent flow through the porous and smooth-wall channels is assumed to be the same; therefore, an amount of the heat required for evaporating the working fluid should be the same in both cases. The power consumption needed for pumping the evaporating heat-transfer agent through the channel is calculated by the formula $N=\Delta P\cdot\dot{m}/\rho$, where ΔP is pressure drop along the channel; $\dot{m}$ is a mass flow rate of the heat-transfer agent, kg/s; ρ is an average density of the two-phase evaporating flow along the channel length. Since the flow rates of the liquids and their thermophysical characteristics are the same in both cases (for porous and smooth-wall evaporation channels), therefore, an energy effect from the use of the porous steam-generating channels can be estimated by the ratio of the differences of the pressure consumption needed for pumping a heat-transfer agent in both channels $k_P=\Delta P_{por}/\Delta P_{sm}$..

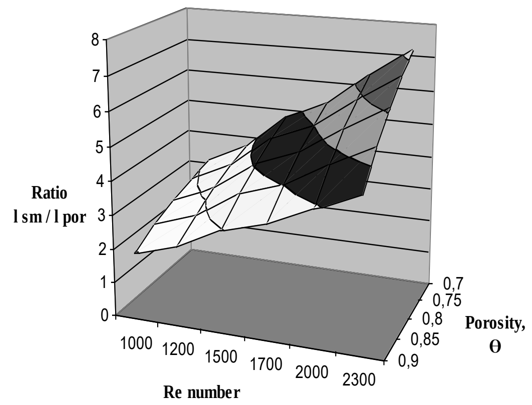

Figure 1 - The value of geometric coefficient of the porous steam-generating channel efficiency $k_F=l_{sm}/l_{por}$ (T_s=360 ^{0}C)

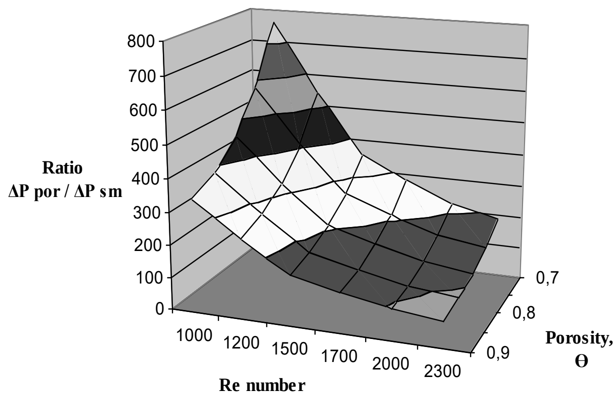

Figure 2 - Ratio of pressure drops in the porous and smooth-wall steam-generating channels $\Delta P_{por}/\Delta P_{sm}$ (T_s=360 ^{0}C)

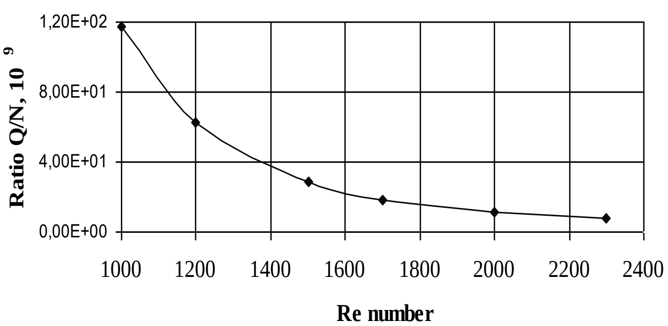

Figure 3 - Q/N ratio, 10^9 for a smooth-wall steam generating channel (T_s=360 ^{0}C)

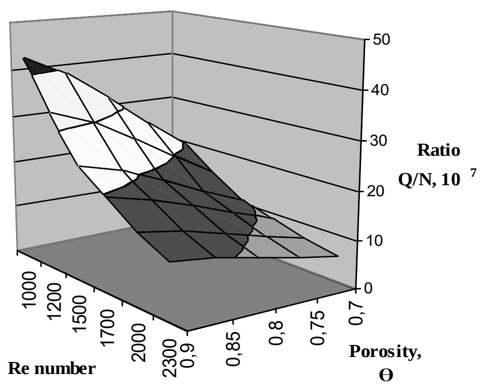

Figure 4 - Q/N ratio, 10^7 for the porous steam generating channel (T_s=360 ^{0}C)

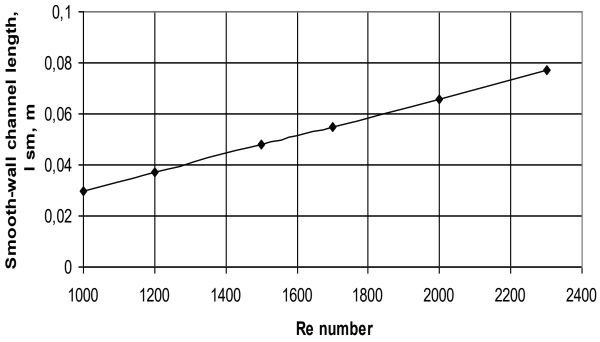

Figure 5 - Length of a smooth-wall steam generating channel l_{sm}, m (T_s=360 ^{0}C)

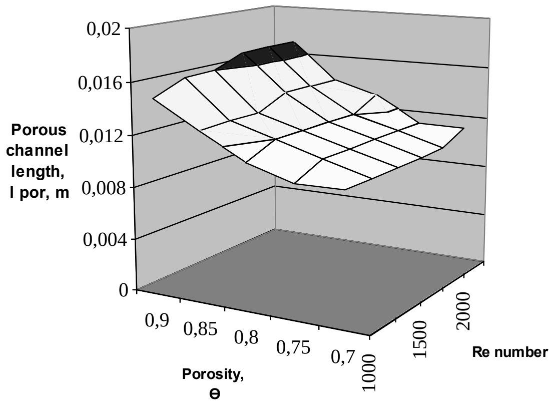

Figure 6 - Length of porous steam generating channel l_{por}, m (T_s=360 ^{0}C)

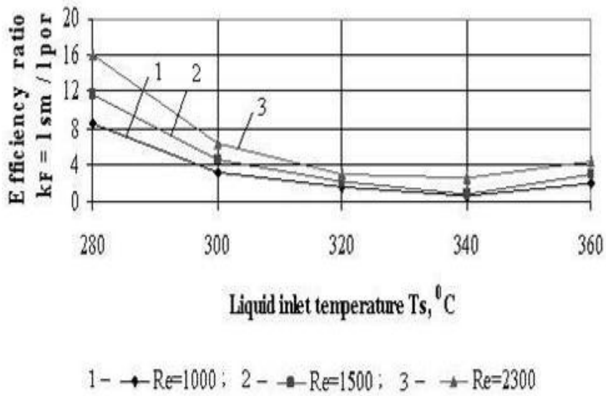

Figure 7 - Dependence $k_F = f(T_s)$ for different Reynolds numbers of the flow at the channel inlet

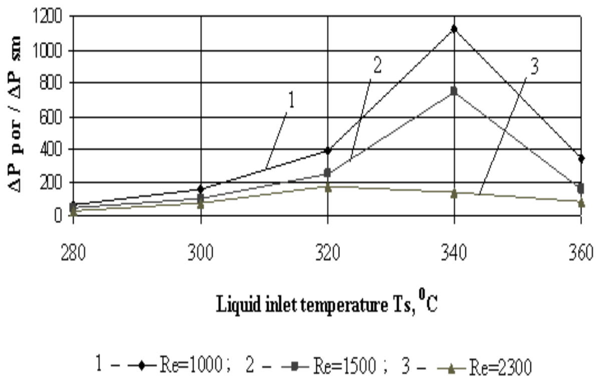

Figure 8 - Dependence $\Delta P_{por} / \Delta P_{sm}$ = f (Ts) for different Reynolds numbers of the flow at the channel inlet

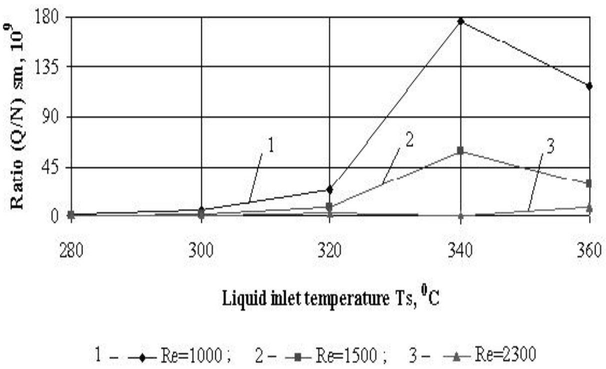

Figure 9 - Dependence Q/N, $10^9 = f(T_s)$ for the smooth-wall channel at different Reynolds numbers of the flow at entry into the channel

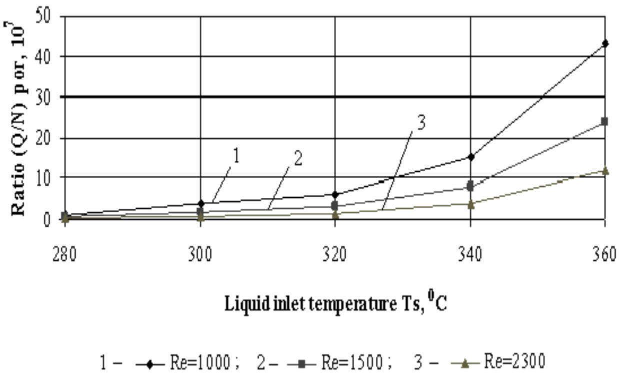

Figure 10 - Dependence Q/N, $10^9 = f(T_s)$ for the porous channel at different Reynolds numbers of the flow at entry into the channel

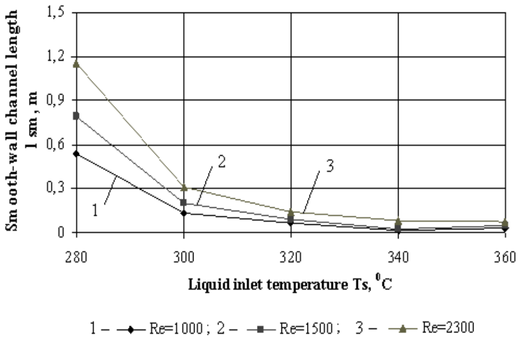

Figure 11 - Dependence l_{sm}=f (T_s) for different values of the Reynolds number of the flow at entry into the channel

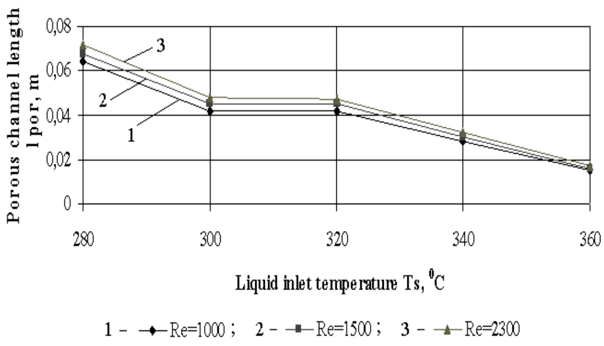

Figure 12 - Dependence l_{por}=f (T_s) for different values of the Reynolds number of the flow at entry into the channel

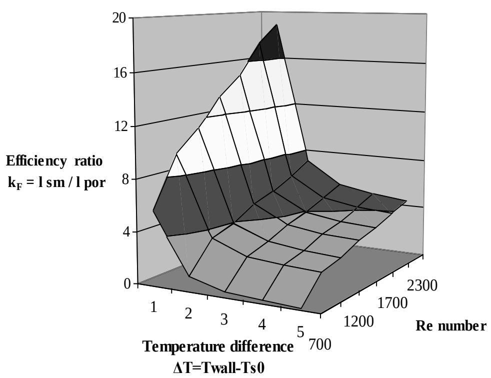

Figure 13 - The value of geometric coefficient of the porous steam-generating channel efficiency
k_F=l_{sm}/l_{por} = *f(Re, ΔT);*
(T_s=360^{0}C*;* T_{wall}=365^{0}C)

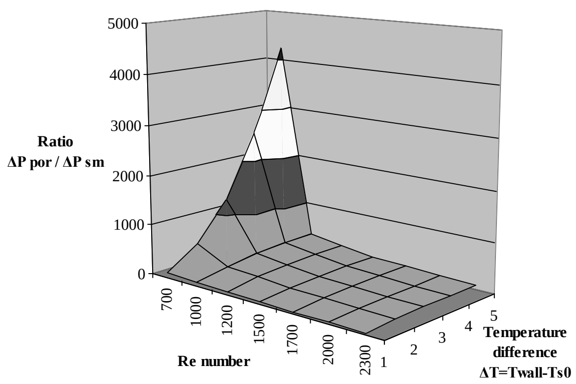

Figure 14 - Ratio of pressure drops in the porous and smooth-wall steam-generating channels
$ΔP_{por}$/$ΔP_{sm}$ = *f(Re, ΔT);*
(T_s=360^{0}C*;* T_{wall}=365^{0}C)

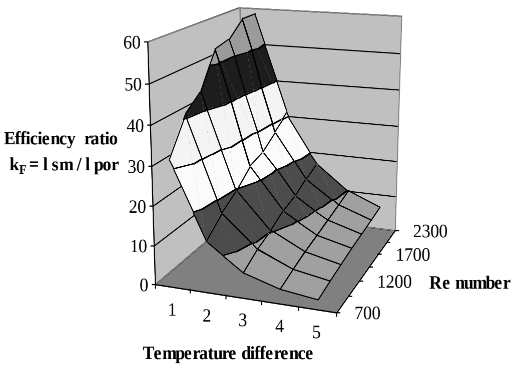

Figure 15 - The value of geometric coefficient of the porous steam-generating channel efficiency
k_F=l_{sm}/l_{por} = *f(Re, ΔT);*
(T_s=*300*^{0}C*;* T_{wall}=*305*^{0}*C*)

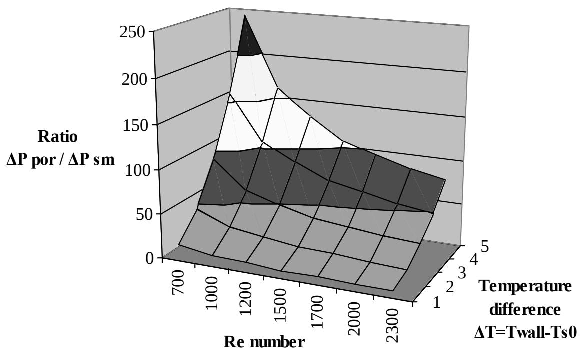

Figure 16 - Ratio of pressure drops in the porous and smooth-wall steam-generating channels
$ΔP_{por}$/$ΔP_{sm}$ = *f(Re, ΔT);*
T_s=300^{0}C*;* T_{wall}=305 ^{0}C)

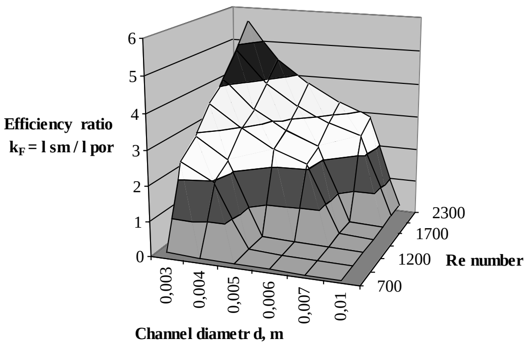

Figure 17 - The value of geometric coefficient of the porous steam-generating channel efficiency
$k_F = l_{sm}/l_{por} = f(Re, \Delta T)$;
($T_s = 360^0$C; $T_{wall} = 365^0$C)

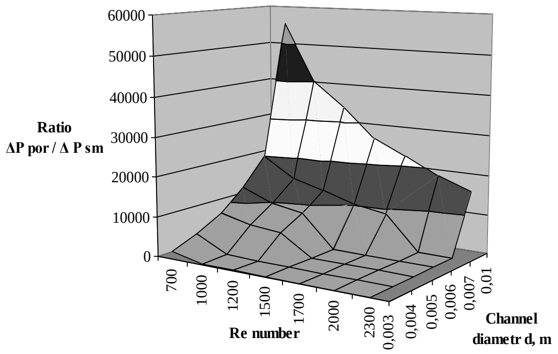

Figure 18 - Ratio of pressure drops in the porous and smooth-wall steam-generating channels
$\Delta P_{por}/\Delta P_{sm} = f(Re, \Delta T)$;
$T_s = 360^0$C; $T_{wall} = 365^0$C)

Conclusions. The above calculations have shown the following:

1. At laminar motion of evaporating liquid and under conditions of the same mass flow rate and the same diameter of channels it is possible to achieve a significant reduction (by several times as much) in length of the porous once-through steam generator in comparison with the length of the smooth-wall once-through steam generator.

2. In case of significantly shorter length of porous once-through steam generators it is not possible under these conditions to reduce pressure drop in them and, consequently, to reduce amount of power consumption needed for pumping coolant.

3. Dependence of geometric coefficient of the porous once-through steam generator efficiency $k_F = l_{sm}/l_{por}$ on the channel porosity and diameter, temperature head and saturation temperature at entry into the channel is inversely proportional; though dependence of this coefficient on the Reynolds number of the flow at entry into the channel is directly proportional.

4. Dependence of the ratio of pressure drops in the porous and smooth-wall steam-generating channels $k_P = \Delta P_{por}/\Delta P_{sm}$ on the porosity and Reynolds number of the flow at entry into the channel is inversely proportional. The dependence of this coefficient on the channel diameter, temperature head and flow saturation temperature at entry into the channel is directly proportional.

5. Despite the fact that in the course of the calculations it was not possible to identify a domain of operating and design parameters of the porous steam generators with water as heat-transfer agent, at which ratio of the pressure difference between the porous and smooth-wall channels is less than 1(due to the significantly shorter length of the porous channel), nevertheless, this study made it possible to establish the main regularities in the behaviour of the energy efficiency coefficients and their dependence on the operating and design parameters of the model. These findings facilitate to specify the main fields for further researches and calculations in order to obtain positive solutions for the problem under consideration.

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About the author

Lukisha Anatolii Petrovych, Candidate of Technical Sciences (Ph.D), Senior Researcher, Senior Researcher in Department of Problems of Mine Energy Complexes, Institute of Geotechnical Mechanics named by N. Poljakov of the National Academy of Sciences of Ukraine (IGTM NAS of Ukraine), Dnipro, Ukraine, lukisha@ukr.net

Про автора

Лукіша Анатолій Петрович, кандидат технічних наук, старший науковий співробітник, старший науковий співробітник у відділі проблем шахтних енергетичних комплексів, Інститут геотехнічної механіки ім. М.С. Полякова Національної академії наук України (ІГТМ НАН України), Дніпро, Україна , lukisha@ukr.net

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Аннотация. В работе представлены результаты расчётов теплогидравлической эффективности пористых прямоточных парогенераторов с модельной рабочей жидкостью вода, в ламинарной области движения теплоносителя и при граничных условиях первого рода. В качестве сравниваемых эталонных поверхностей использовались гладкостенные цилиндрические каналы различных диаметров. В качестве расчетной базы были взяты следующие режимно-конструктивные параметры: температура жидкости на линии насыщения на входе в канал T_{so}=280 °С, 300 °С, 320 °С, 340 °С, 360 °С. Температурный напор – разность между температурой стенки и температурой жидкости на входе $\Delta T=T_w-T_o$ = 1, 2, 3, 4, 5 °С. Число Рейнольдса на входе в канал Re_o = 700, 1000, 1200, 1500, 1700, 2000, 2300. Пористость канала θ = 0,7; 0,75; 0,8; 0,85; 0,9. Пористый материал – металловойлок с диаметром медных волокон 200 мкм. Диаметр канала d =$3\cdot10^{-3}$; $4\cdot10^{-3}$; $5\cdot10^{-3}$; $6\cdot10^{-3}$; $7\cdot10^{-3}$; $10\cdot10^{-3}$ м . На основании проведенных расчётных исследований был сделан вывод, что для условий одинаковых массовых расходов охладителя, при ламинарном режиме движения, и одинаковых диаметрах каналов, можно добиться существенного сокращения длины пористого прямоточного парогенератора по сравнению с гладкостенным прямоточным парогенератором, однако в данных условиях не удаётся достичь меньших значений перепада давления в них и соответственно меньших мощностей, затрачиваемых на прокачку теплоносителя. Данное расчётное исследование так же позволило выявить основные закономерности в поведении энергетических коэффициентов эффективности и их зависимость от режимно-конструктивных параметров модели.

Ключевые слова: теплогидравлическая эффективность; пористые парогенерирующие каналы; теплоноситель вода; граничные условия первого рода; ламинарный режим движения теплоносителя.

Анотація. В роботі представлені результати розрахунків теплогідравлічної ефективності пористих прямоточних парогенераторів з модельною робочої рідиною вода, в ламінарної області руху теплоносія і при граничних умовах першого роду. В якості порівнюваних еталонних поверхонь використовувалися гладкостінні циліндричні канали різних діаметрів. В якості розрахункової бази були взяті такі режимно-конструктивні параметри: температура рідини на лінії насичення на вході в канал T_{so}=280 °С, 300 °С, 320 °С, 340 °С, 360 °С. Температурний напір - різниця між температурою стінки і температурою рідини на вході $\Delta T=T_w-T_o$ = 1, 2, 3, 4, 5 °С. Число Рейнольдса на вході в канал Re_o = 700, 1000, 1200, 1500, 1700, 2000, 2300. Пористість каналу θ = 0,7; 0,75; 0,8; 0,85; 0,9. Пористий матеріал - металовойлок з діаметром мідних волокон 200 мкм. Діаметр каналу d=$3\cdot10^{-3}$; $4\cdot10^{-3}$; $5\cdot10^{-3}$; $6\cdot10^{-3}$; $7\cdot10^{-3}$; $10\cdot10^{-3}$ м. На підставі проведених розрахункових досліджень був зроблений висновок, що для умов однакових масових витрат охолоджувача, при ламінарному режимі руху, і однакових діаметрах каналів, можна домогтися істотного скорочення довжини пористого прямоточного парогенератора в порівнянні з гладкостінним прямоточним парогенератором, проте в даних умовах не вдається досягти менших значень перепаду тиску в них і відповідно менших потужностей, що витрачаються на прокачування теплоносія. Дане розрахункове дослідження також дозволило виявити основні закономірності в поведінці енергетичних коефіцієнтів ефективності та їх залежність від режимно-конструктивних параметрів моделі.

Ключові слова: теплогідравлічна ефективність; пористі парогенеруючі канали; теплоносій вода; граничні умови першого роду; ламінарний режим руху теплоносія.

Стаття надійшла до редакції 25.02.2021